AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 44 No. 7 JULY 1954 PRICE: 3s. 6d.

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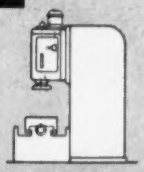
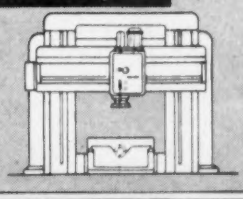
HELLER PLANO-TYPE MILLING MACHINES

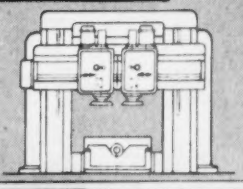
TYPE SFS



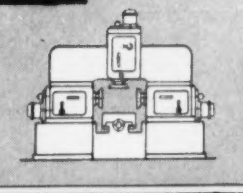
TYPE LFS 1-spindle



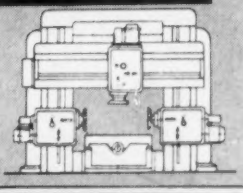
TYPE LFS 2-spindle



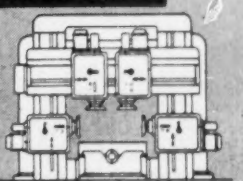
TYPE SFK



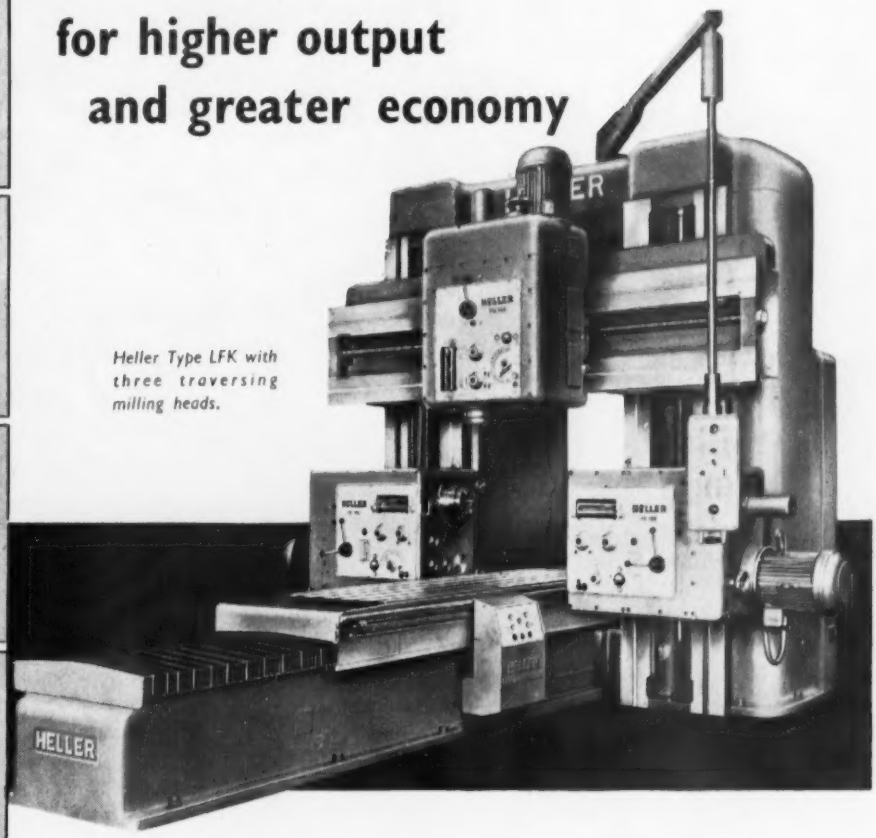
TYPE LFK 2 or 3-spindle



TYPE LFK 4-spindle



for higher output and greater economy



Heller Type LFK with three traversing milling heads.

The simultaneous milling of several faces on one workpiece—often in different planes and at different angles—is the answer to the demand for increased output of many complex parts. It obviates re-clamping and re-aligning and in many cases reduces the number of machines required for production.

Heller Plano-type Milling Machines are designed on the unit-assembly system and can be literally 'built around the component'. The major units are the milling heads, tables, feed mechanism, uprights and frames. The selection of units depends entirely on the work required; for example, light milling heads and a large table can be used for bulky workpieces with only small faces to be milled, or alternatively, heavy milling heads can be employed with a small table for heavy milling on small components. The power employed is therefore related to the workpiece and not to machine table size, and milling heads of different sizes can be fitted to the same machine. The economic advantages of this system of construction are therefore extended into actual operating, saving power and floor space.

Some of the most commonly used arrangements are shown opposite. There is a catalogue available which gives full details. Write for a copy now.

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On June 15th 1215—in face of a vast assembly amongst which he had hardly a single friend—King John sealed the famous Magna Carta at Runnymede, close to Windsor. A long document of 63 clauses, some are but trifling concessions to individuals whilst others are of primary importance in the constitutional history of England. Four copies of Magna Carta sealed with the great seal of King John are in existence today. Two are in the British Museum, the others belong to the cathedrals of Lincoln and Salisbury.

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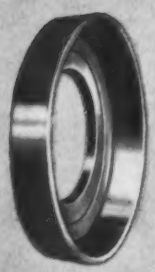

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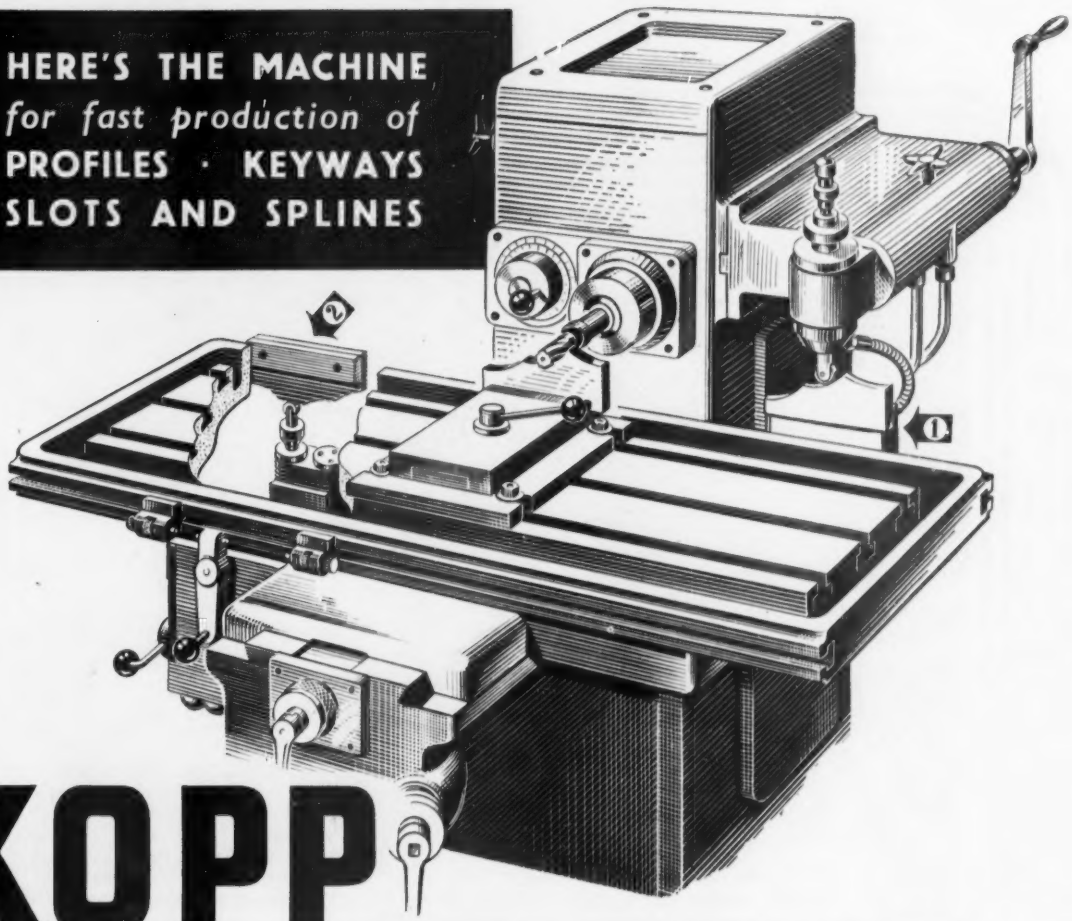
HERE'S THE MACHINE
for fast production of
PROFILES · KEYWAYS
SLOTS AND SPLINES

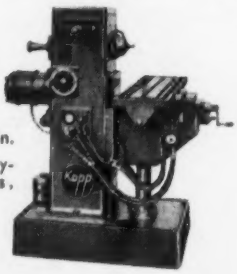


KOPP

Hydraulic MILLING MACHINE

This fully hydraulic horizontal milling machine produces keyways, slots, splines, etc., and incorporates a special arrangement for rapid profiling lengthways. A simpler type is available for milling straightforward keyways, slots and splines only.

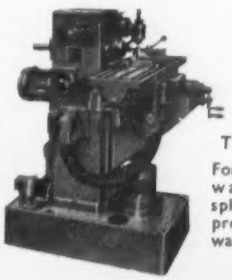
The profile machine shown above incorporates hydraulic feeler controls for both vertical and longitudinal travel of the table. Cam (1) controls vertical movement in accordance with the profile of the work, and cam (2) controls the speed of the table. This control of table speed is an important advantage ; feed of the work table need not be governed by the most unfavourable section of the profile, and uniform feed can be maintained in keeping with the profile of the work. This results in faster output and improved finish. Feed of the cutter spindle to and from the work and traverse of the work table are accomplished by hydraulic power.



TYPE ENF o/n.
For milling keyways, slots, splines, etc.

CAPACITIES

Width of grooves milled	up to 25 mm.
Stroke of table (length milled)	up to 150 mm.
Depth of cut	up to 50 mm.
Table space available for work	200 × 900 mm.



TYPE EKF O.
For milling keyways, slots, splines and for profiling lengthways.

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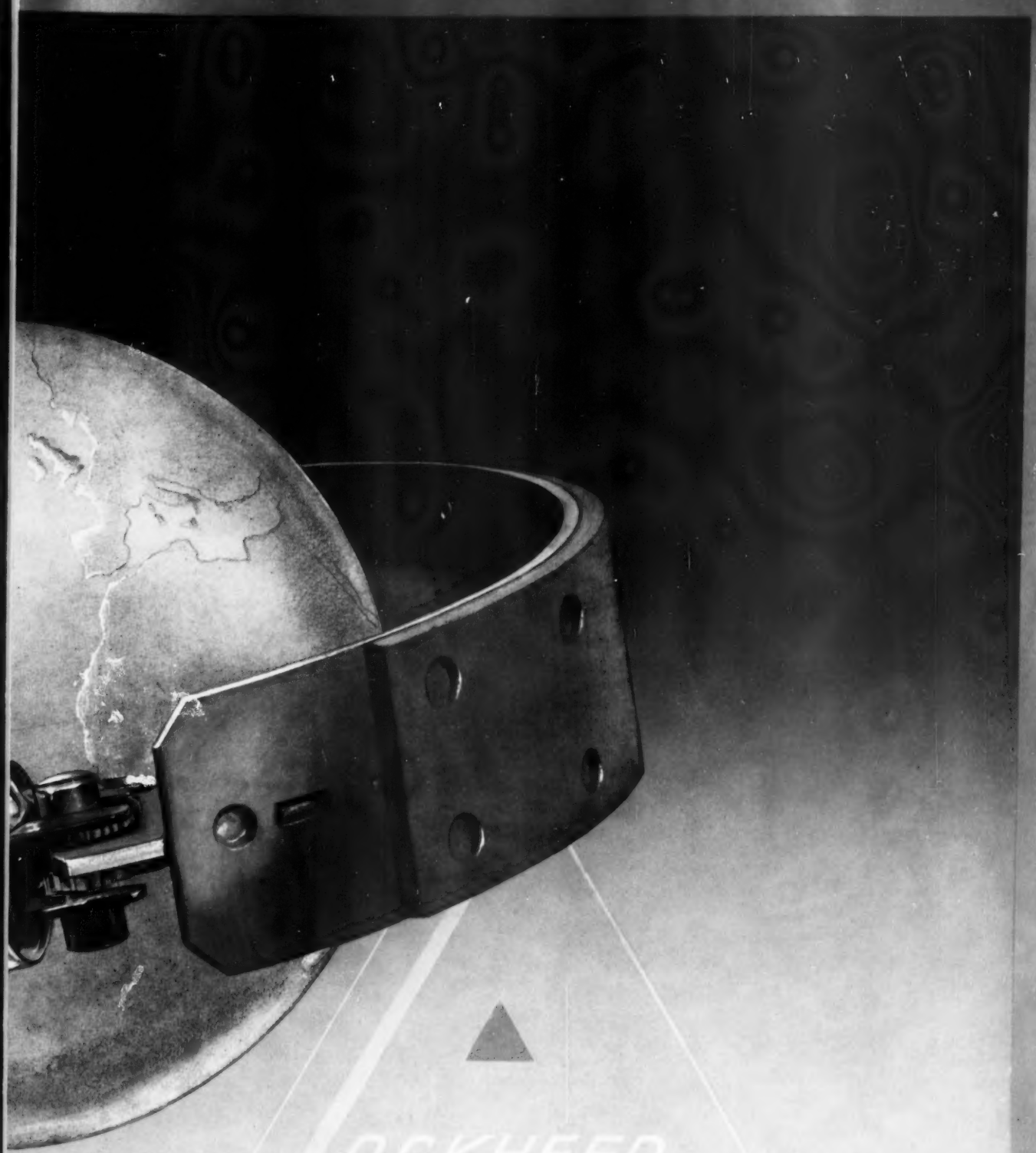

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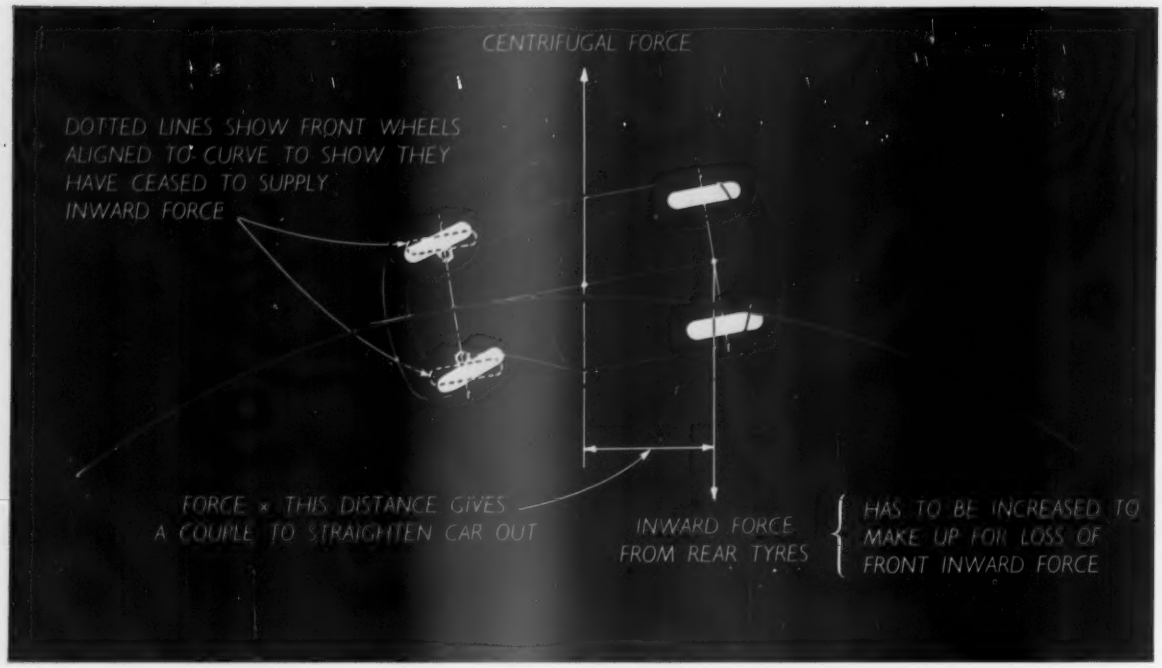
CENTRIFUGAL FORCE
DOTTED LINES SHOW FRONT WHEELS ALIGNED TO CURVE TO SHOW THEY HAVE CEASED TO SUPPLY INWARD FORCE
FORCE × THIS DISTANCE GIVES A COUPLE TO STRAIGHTEN CAR OUT
INWARD FORCE FROM REAR TYRES
HAS TO BE INCREASED TO MAKE UP FOR LOSS OF FRONT INWARD FORCE

SUBTLETIES OF STEERING

The Transient Stage:
3. Entering and Leaving a Bend

During steady cornering conditions only a part of the outward force resisted by the front wheels is caused by the front end mass of the vehicle, trying to fly outward on the bend. The rest is due to the combined effects of drag distribution, under the dissimilar weights carried by inner and outer wheels, the self-righting or straightening torque of the wheels which exists while they are drifting, and camber thrust, due to the wheels leaning outward. All of these depend upon the tyre tread elements having settled down into a steady pattern of behaviour, and we have already seen, in discussing the results of Kenneth Stonex's paper, that a little time (or more correctly distance) is required to arrive at the steady tread pattern, and hence the steady values of these effects. During the transient stage, in entering or leaving a bend, their values will therefore tend to lag behind what would be expected in steady conditions.

While a car is entering a bend a higher load is required at the front wheels than is called for to produce the centripetal accelerations of the front end mass, because a rotational acceleration must also be given to the vehicle. This tends to compensate for the lagging tyre tread effects and the car behaviour is roughly comparable to its normal steady cornering behaviour.

When a car is leaving a bend a different state of affairs exists. Front wheel steering angle has to be let off so that centrifugal force can provide the necessary torque to destroy the rotational velocity of the vehicle round its centre of gravity. Drift angle of the front tyres is either eliminated, or in cases where steering angle is let off very rapidly, may even be reversed in direction. There is a corresponding increase in the inward force which must be supplied by the rear tyres, and hence in the rear drift angle. For normal and relatively unhurried cornering, where the rear tyre has a considerable untapped margin of cornering force, this does not matter, and car behaviour can be normal and easily controlled.

Even when cornering near the limit with an under-steering car there is still a margin of cornering force on the rear tyres, and the behaviour may still be reasonable. When cornering near the limit with a neutral or over-steering car the rear wheels may be pushed over the edge into skidding, with any attempt to get out of the corner quickly, and control of the car will cease to be as definite as could be desired.

In the ordinary way, in leaving a bend, a good deal of margin for variability exists and the driver, by taking his time over the manœuvre, retains control even in adverse circumstances such as those listed above. This is often not the case, however, when transferring from the first to the second part of an S-bend, where timing must be precise if the vehicle is to be correctly placed for the second part, and where the change may have to be rapid because of the road layout. The over-steering car may momentarily skid, which is usually surprising and can be disconcerting.

It is worth noting here that the under-steering car in such conditions has a large front wheel drift and fairly large changes of front wheel steering angle are required to cause appreciable changes in sideways force. The exact front wheel angle is therefore not critical and control is relatively simple. An over-steering car in such circumstances is much more sensitive to front wheel steering angle.

We see that leaving a corner is a more difficult and more hazardous operation than entering it.

Thompson
Self-adjusting
STEERING ROD ASSEMBLY
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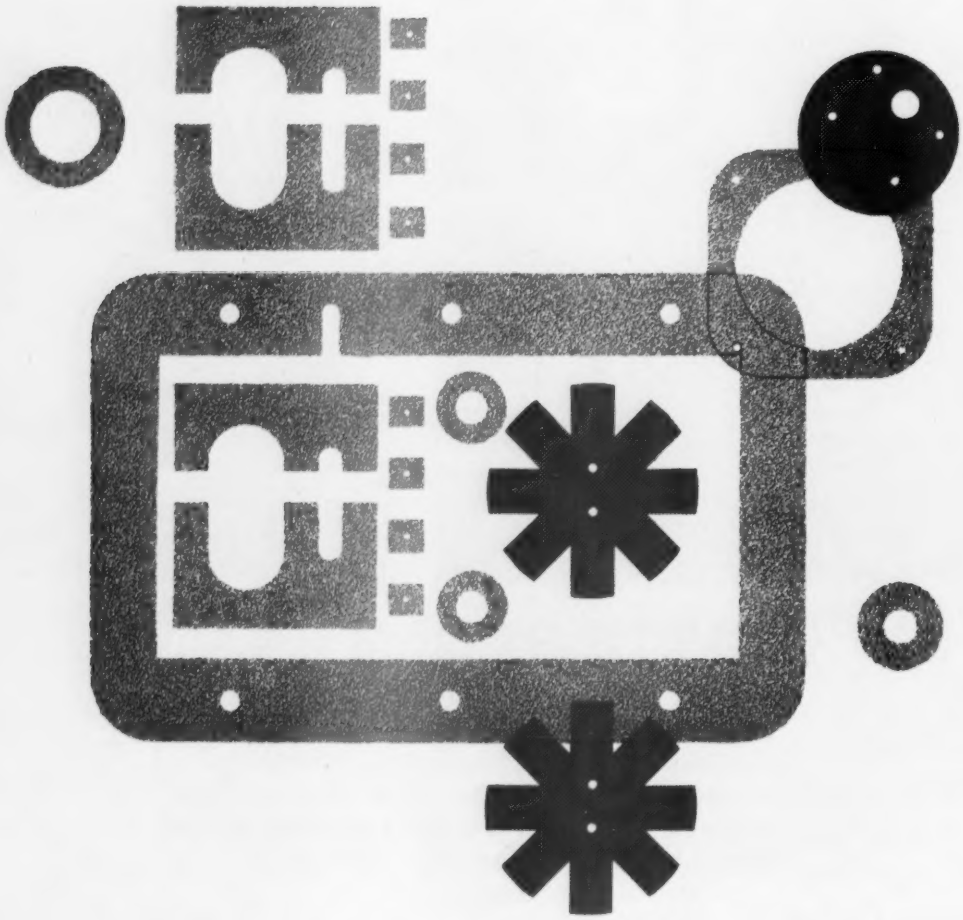

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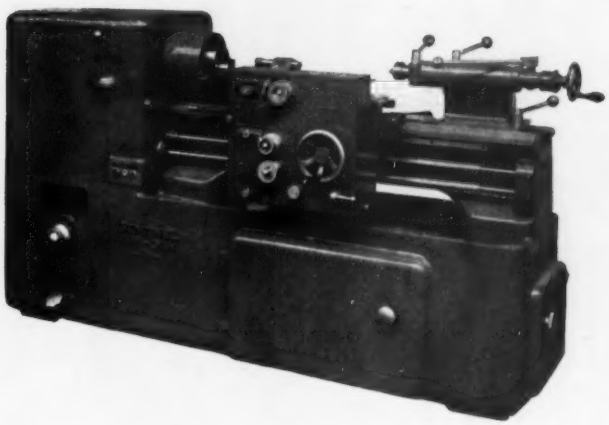
Maximum thread diameter 4″ (external), 6″ (internal); 3′ 0″ admitted between centres.



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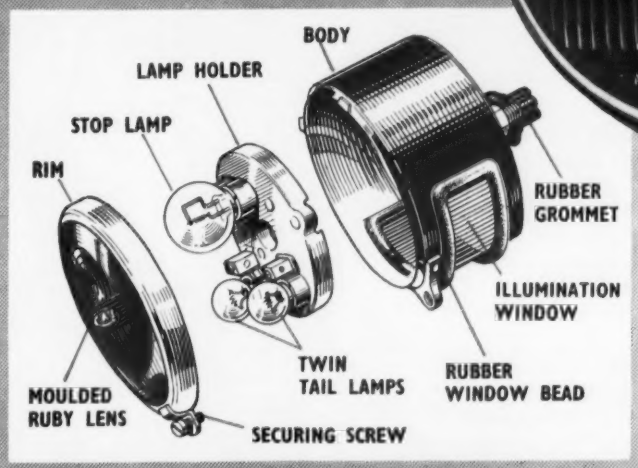

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TYPE VP 56

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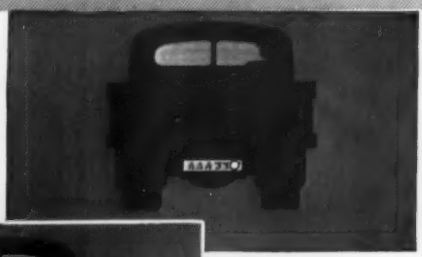
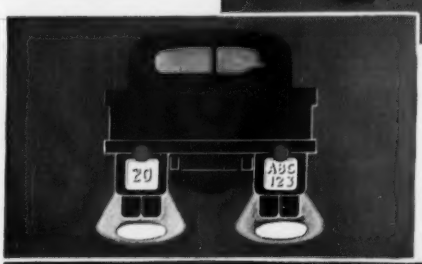
BELOW: Twin C.A.V. lamps, extra large diameters, unmistakable visibility.





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Fuel Injection and Electrical Equipment

C. A. V. LIMITED, ACTON, LONDON, W. 3

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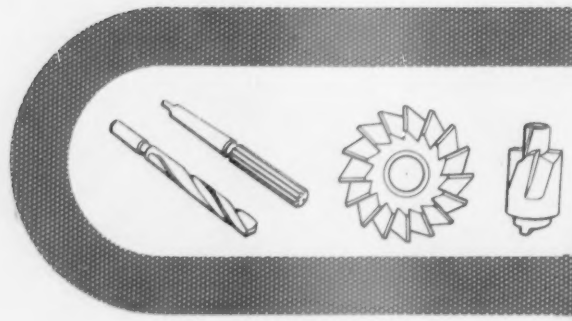

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for the Small Tool Industry

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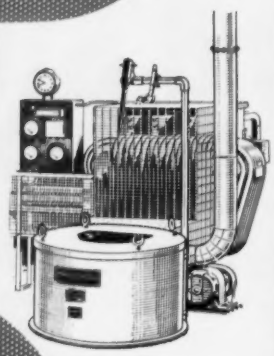

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CLAYTON DEWANDRE COMPANY LIMITED LINCOLN ENGLAND

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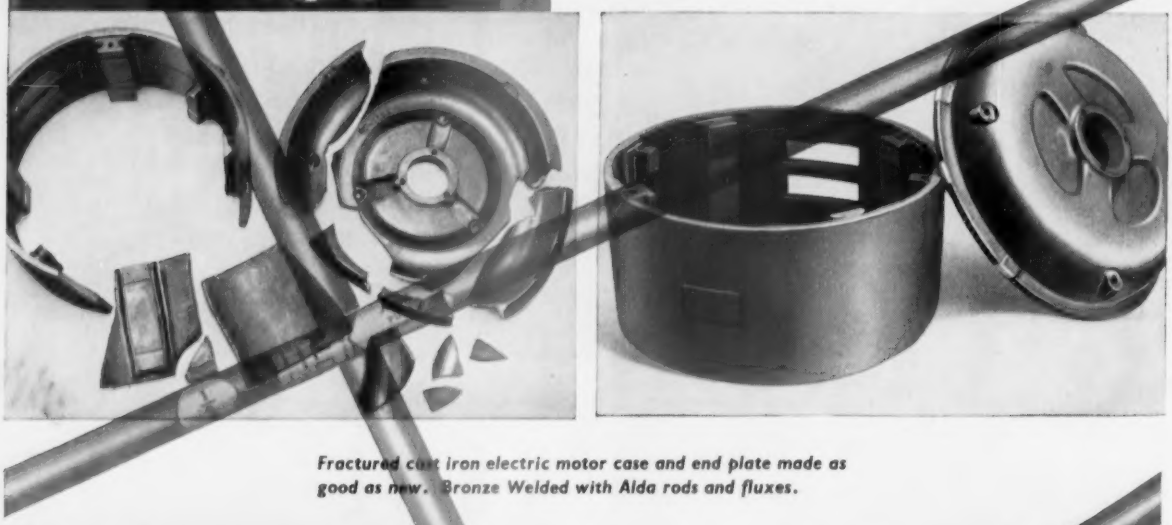
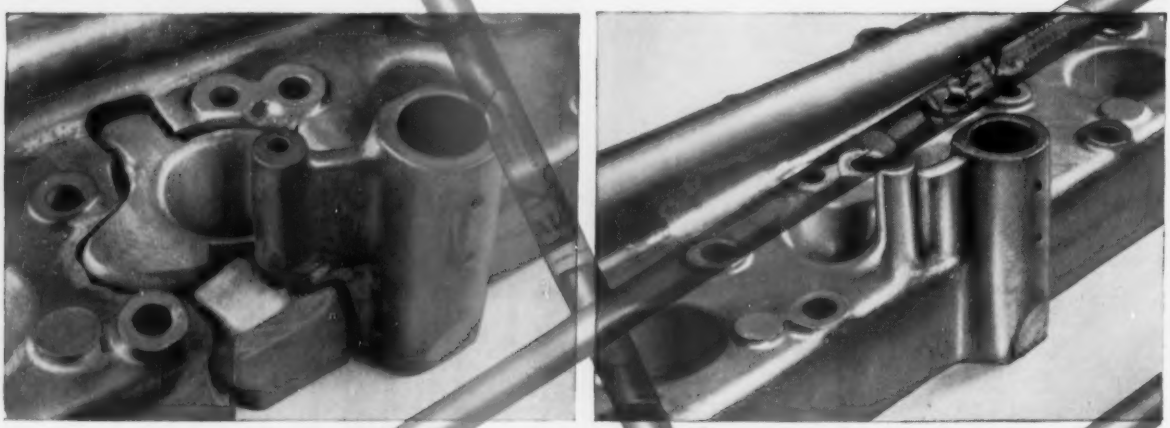


Fractured cast iron electric motor case and end plate made as good as new. Bronze Welded with Alda rods and fluxes.



A fractured aluminium cylinder head before and after repair with Alda rods and fluxes. At left, the casting is shown prepared for welding, and at right, the finished job — as good as new.

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This cast-iron binder attachment is being repaired by welding a lug on to the arm. The Blowpipe is the B.O.C. high pressure type CH, the rod an Alda Cast Iron Super Silicon, and the flux Alda cast iron.

Gas welding is widely used for repair and reconditioning in the automobile industry — for aluminium and sheet steel bodies as well as aluminium and cast iron engine parts. Here, a shattered wing is being made good, using an Alda Mild Steel Copper coated welding rod.

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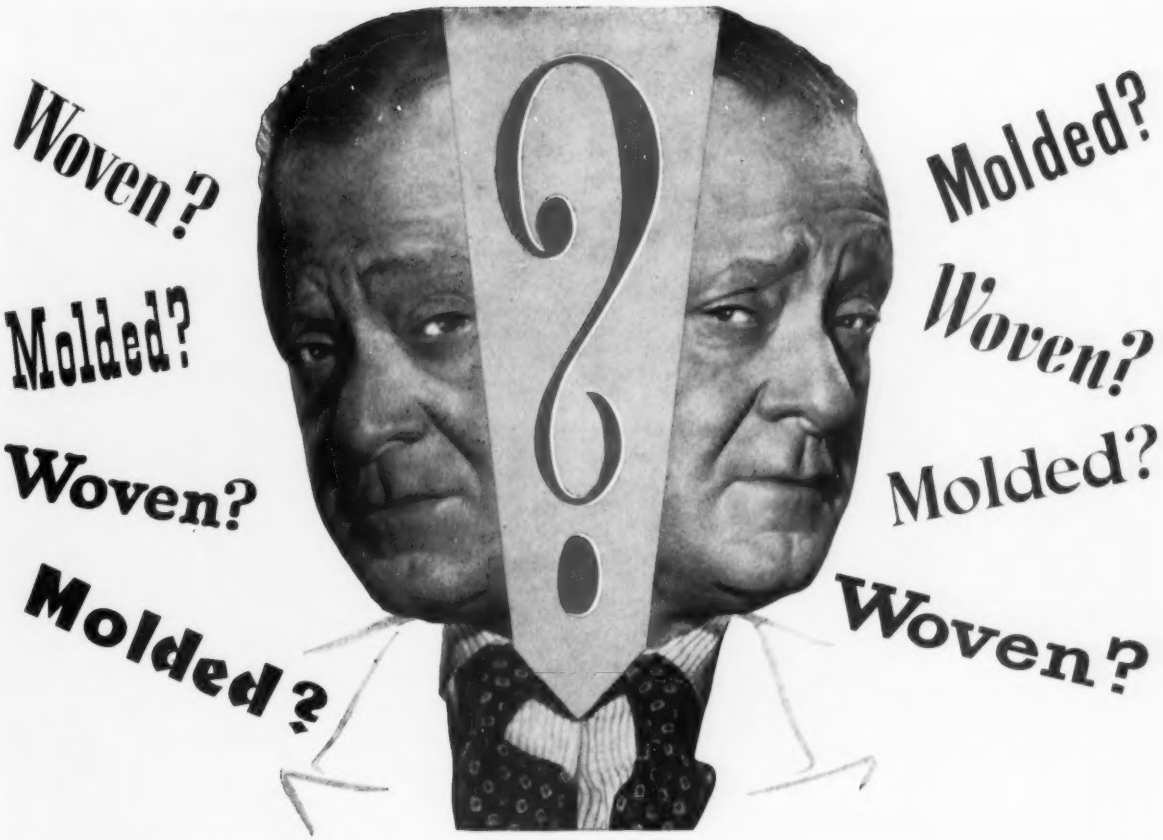

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DB 21

TOOLROOM FAVOURITES

Illustration by courtesy of Siemens Brothers & Co. Ltd.

KE 970 This type of 2% Carbon 14% Chrome Steel is recommended for the highest duty work possessing the desired properties of deep hardening, a high resistance to abrasion together with the minimum amount of distortion in hardening. It also has the ability to withstand drawing in tempering after hardening over a much wider range of temperature than is normally permissible with standard alloy tool steel types. We also supply K. E. 961 1½% Carbon 14% Chrome Steel in cases where increased toughness is desired at the expense of a slight reduction in hardness.

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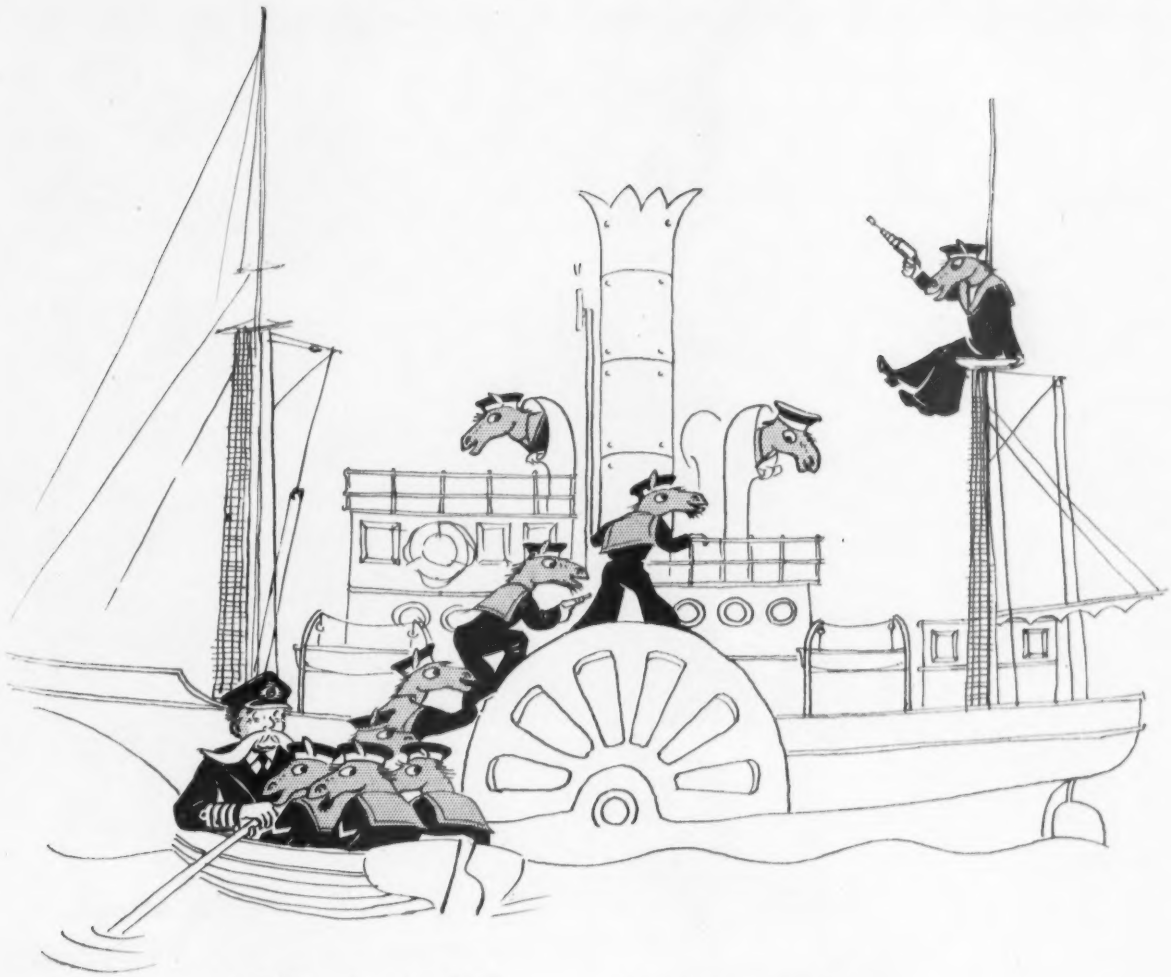

ALL I ASK — *said the Managing Director . . .*
. . . the Shipbuilder's Friend—

is a tall ship or a short ship or a thin or a fat ship—*any* sort of ship so long as she is off the slipways and has power aboard.

Let me put a team of my Little Horses aboard such a ship — said the Managing Director — and before you can say Ben Bolt they'll be drilling holes and screwing screws and fixing nuts so fast that she'll be ready to sail before the crew has got aboard the train at Euston. Show a leg, me hearties, Desoutter's aboard!

Desoutter tools *put you all at sea*

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CRC 256

Desert Grit

The fleet of Thornycroft "Mighty Antars" used by the Iraq Petroleum Company to lay a pipeline to the Mediterranean Coast were fitted with Hardy Spicer propeller shafts and universal joints. To carry such immense loads over desert roads, travelling perpetually in a cloud of dust and grit, would tax the strength of the toughest vehicle. Hardy Spicer —specialists in motor transmission, have designed shafts and couplings which send racing cars hurtling round tracks at 150 m.p.h., drive coaches and lorries up mountain passes, and now, their latest achievement, to keep a fleet of heavily laden "Mighty Antars" rumbling across the desert.

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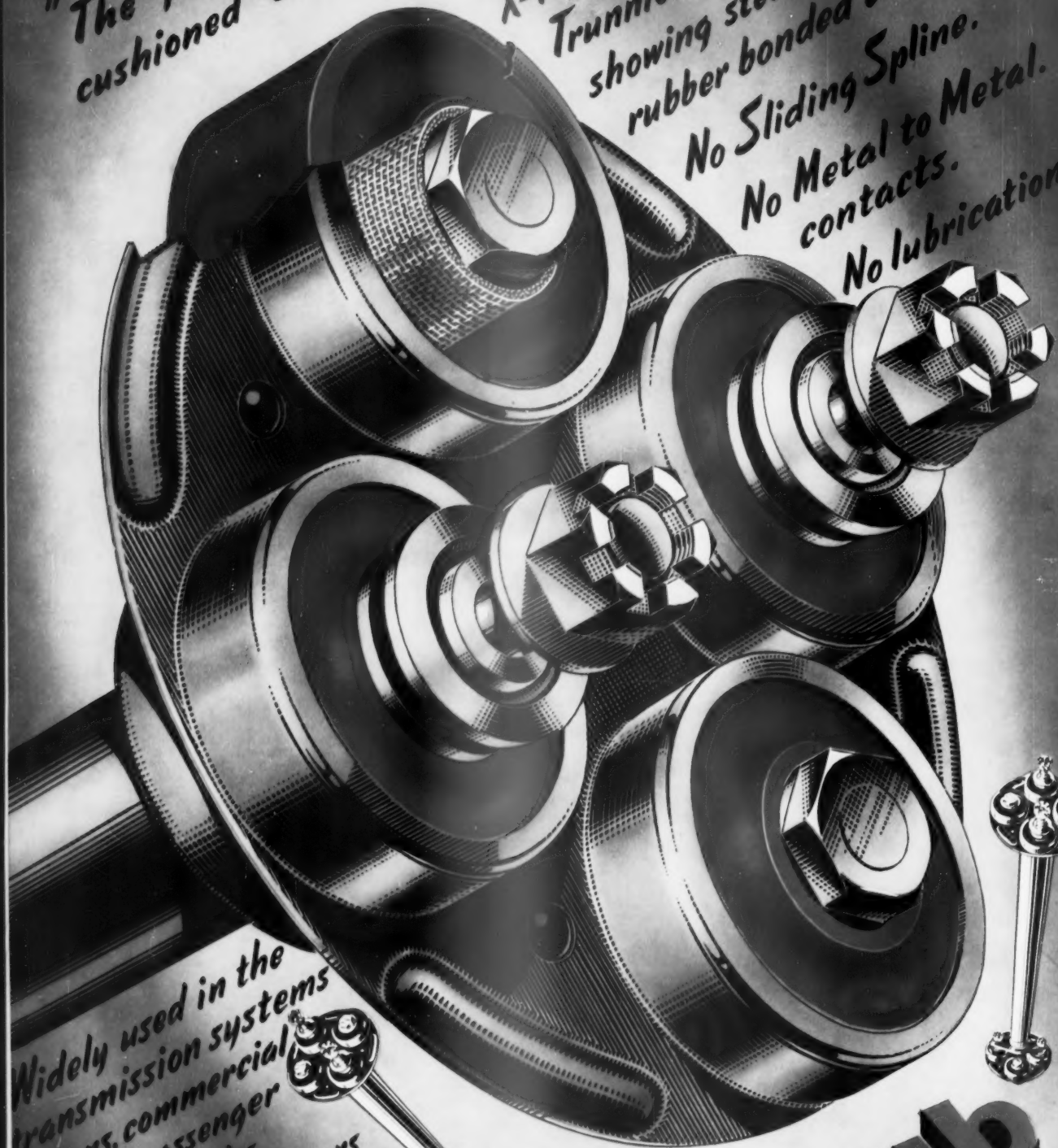

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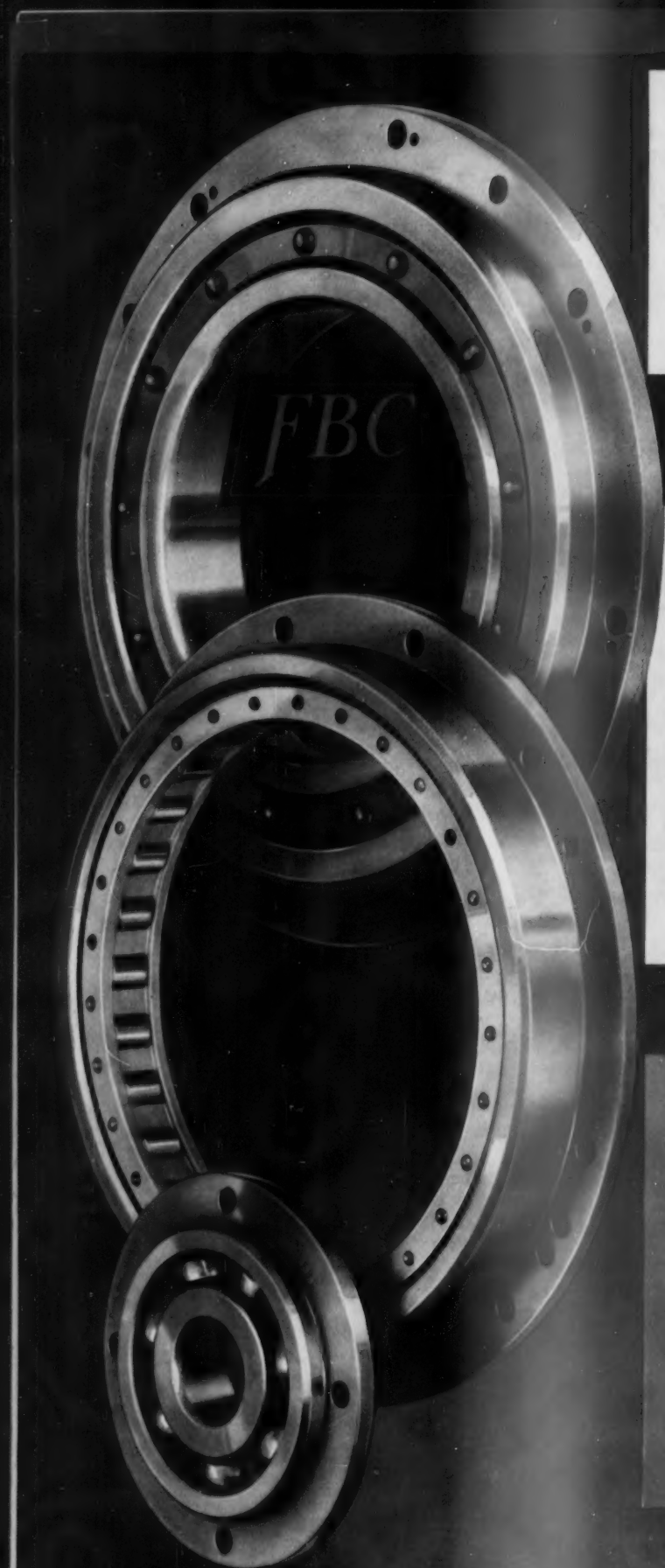
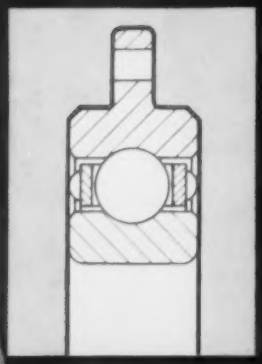
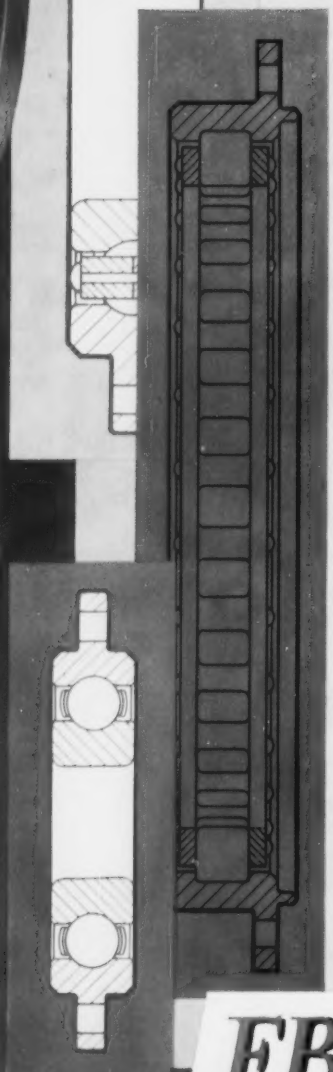




SPECIALS



There are many places in which F.B.C. 'specials' are a great aid to the designer— they can save space and weight, carry greater loads in the same space, or run at unusually high speeds or stand up to unpleasantly high temperatures and in many ways benefit the design and performance.

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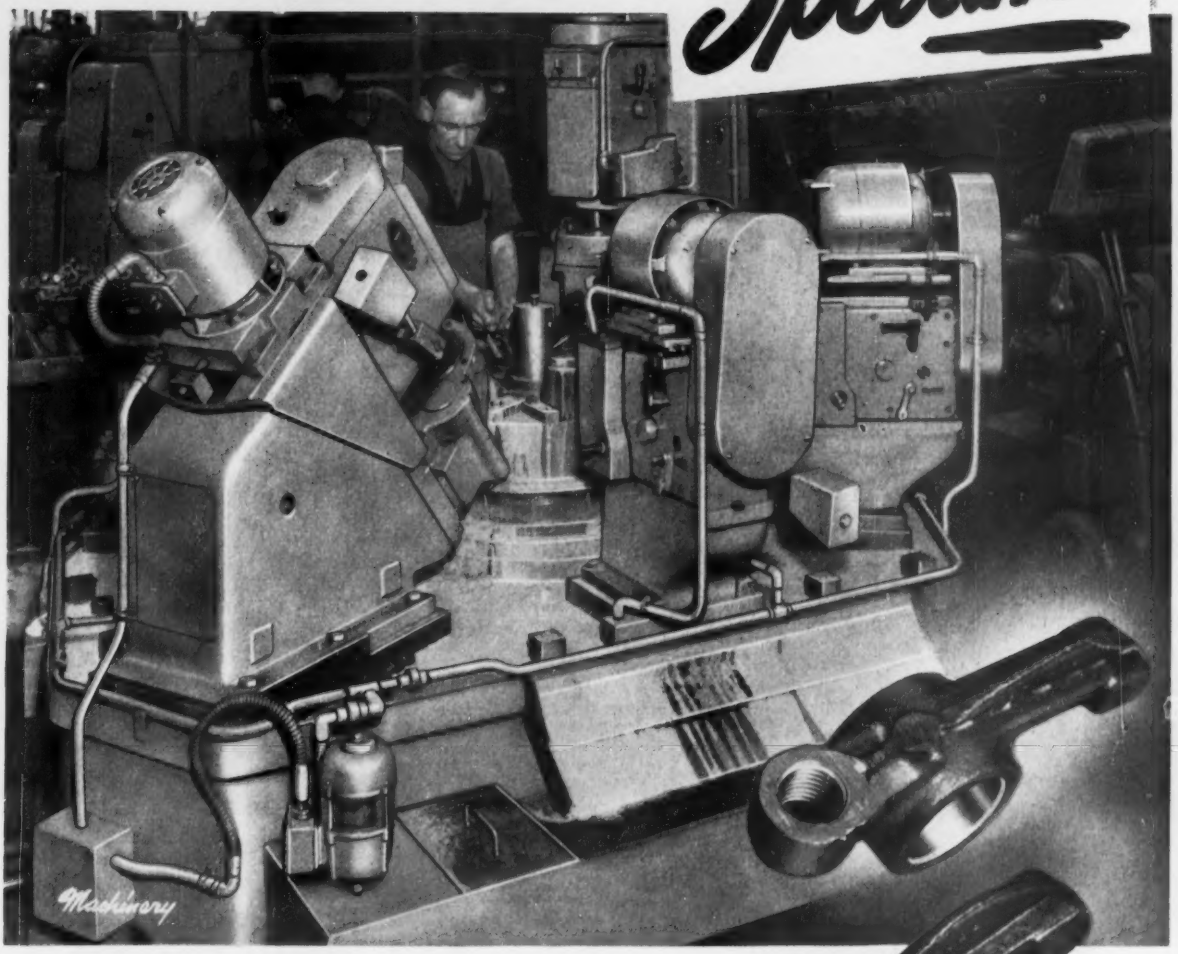

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This 6-station machine for milling, drilling, countersinking and tapping valve rockers, is typical of hundreds of *ARCHDALE* specials supplied to the automobile industry.

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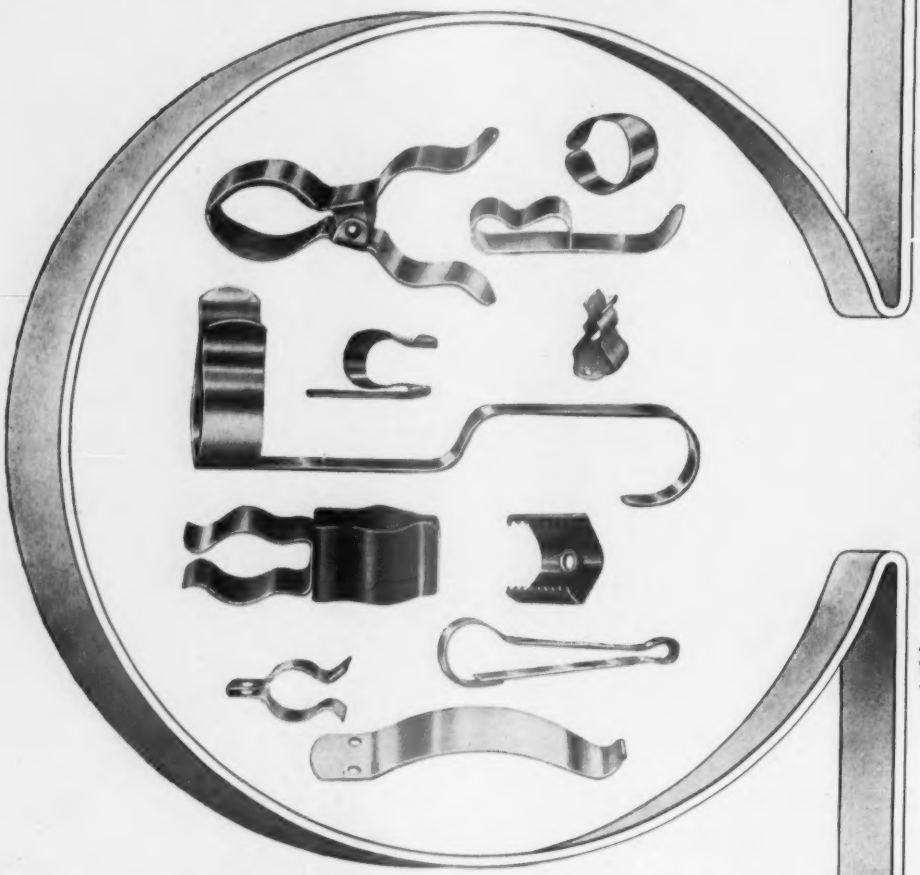

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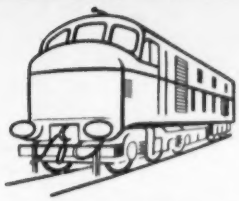
HT6A

THINK ZINC



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The Association welcomes inquiries about the use of zinc alloy die castings. Publications and a list of Members are available on request. We suggest you write for our booklet " Zinc Alloy Die Castings and Productivity."

ZINC IS NOW PLENTIFUL. THERE ARE NO RESTRICTIONS ON ITS USE.

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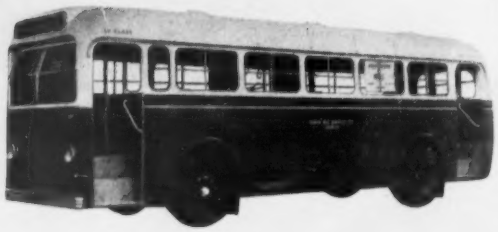

In Kenya they ride with 'MG 5'

REGD. TRADE MARK

The interior of this bus, built for service in Kenya, reveals the workmanship and finish that distinguishes all bodies by the famous firm of Duple Motor Bodies Limited, of Hendon. The chequer-plate flooring, illustrated, is of James Booth 'MG5' aluminium alloy, and this alloy in sheet form is used for roof lining and exterior panels.

Only one-third the weight of steel, 'MG5' saves fuel and tyre wear yet it can be relied upon to give long service under the most arduous conditions because of its resistance to corrosion and road shocks.

Our Development Department will gladly give information on the use of James Booth alloys in road transport, entirely without obligation.

BOOTH BIRMINGHAM

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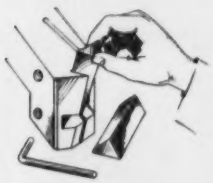
DP.1

Wimet CLAMPTIP TOOLS offer even longer life from carbide tools

The Wimet range of Clamptip tools offers users the benefits of improved tool performance, infinitely longer carbide life, and a variety of tooling possibilities.
These tools may be used in any combination one with another or incorporated with standard tools in a set-up.

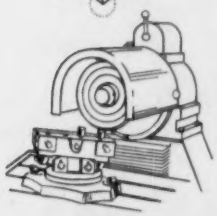


STRESS-FREE SOLID INSERTS

Tool performance is improved since the solid carbide inserts are secured entirely by mechanical means and thus remain free from the stresses usually inherent in brazed tools.

MULTIPLE CUTTING POINTS

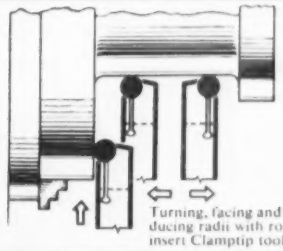
When cutting edges become dulled, the insert can be repositioned in its holder. Only when all edges have been so used is it necessary to remove for resharpening.



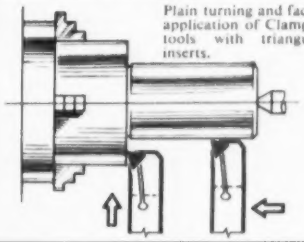
SIMPLIFIED RESERVICING AND UNDISTURBED TOOL SETTINGS

In multi-tool set-ups the use of Clamptip tools enables tool settings to remain undisturbed when carbide inserts require removal for reservicing. At the end of the long carbide life of the tool, the insert is replaced with another of standard size (alternatively, the remaining unused carbide may be brazed to a steel insert when the fullest possible service may then be obtained from the Wimet).

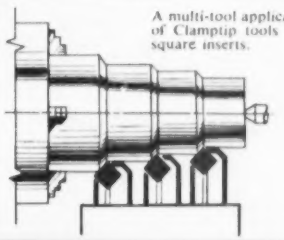
Typical applications



Turning, facing and producing radii with round-insert Clamptip tools.



Plain turning and facing application of Clamptip tools with triangular inserts.



A multi-tool application of Clamptip tools with square inserts.

Made in Three Types

Wimet Clamptip Tools are made with round, square, or triangular inserts in a range of sizes of each type. Inserts are available in grades of Wimet recommended for roughing or finishing cuts on either cast iron and non-ferrous materials or steels.

Write to-day for your copy of the Wimet Clamptip Tools leaflet. It contains full details of all types, sizes, and grades of Wimet inserts available.

WICKMAN *of* COVENTRY

WICKMAN LIMITED, WIMET DIVISION, TORRINGTON AVENUE
COVENTRY, ENGLAND. *Telephone: Tile Hill 66621*

Wickman

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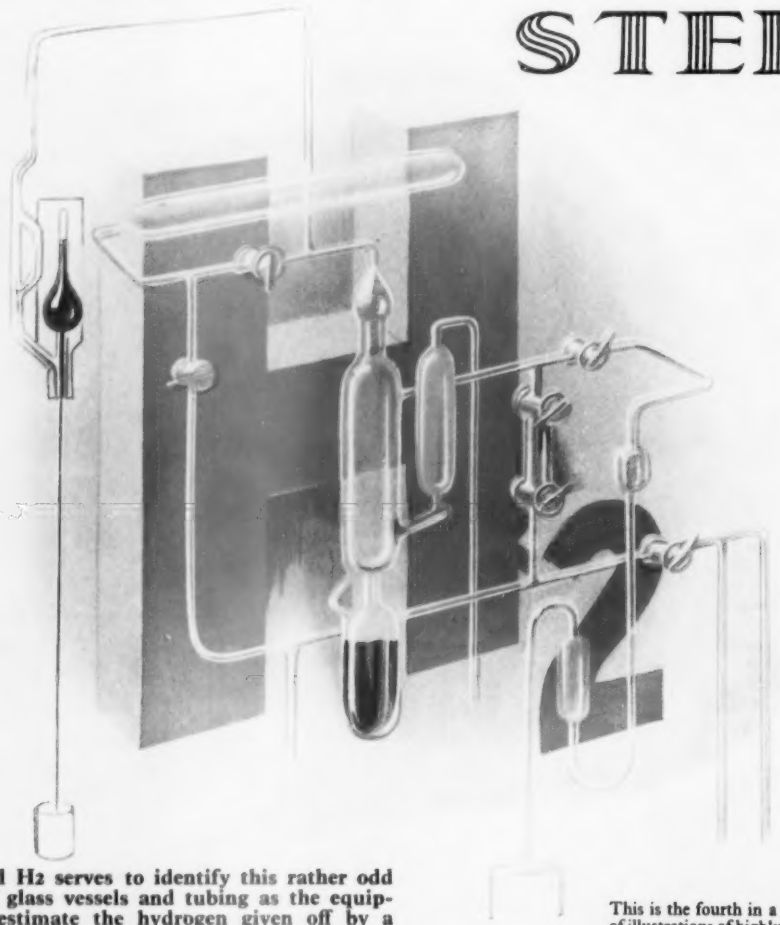
*Formula for*_____

STEEL



The Symbol H_2 serves to identify this rather odd arrangement of glass vessels and tubing as the equipment used to estimate the hydrogen given off by a heated specimen of steel.

To the steelmaker, hydrogen is a "gremlin" or trouble-maker. Absorbed during melting, it may be released on solidification causing "blown" ingots particularly with stainless or high-speed steels. Or it may be retained until the steel is manipulated as in forging when it may come out of solid solution and give rise to internal cracks.

Unremitting routine checks on bath samples offer the only means of detecting high hydrogen contents in good time so that the melts may be either rejected or effectively treated.

This is the fourth in a series of illustrations of highly specialised equipment devised to control the production and testing of special steels by
William Jessop & Sons Ltd.
(As and when the illustrations appear, copies will be supplied on request).

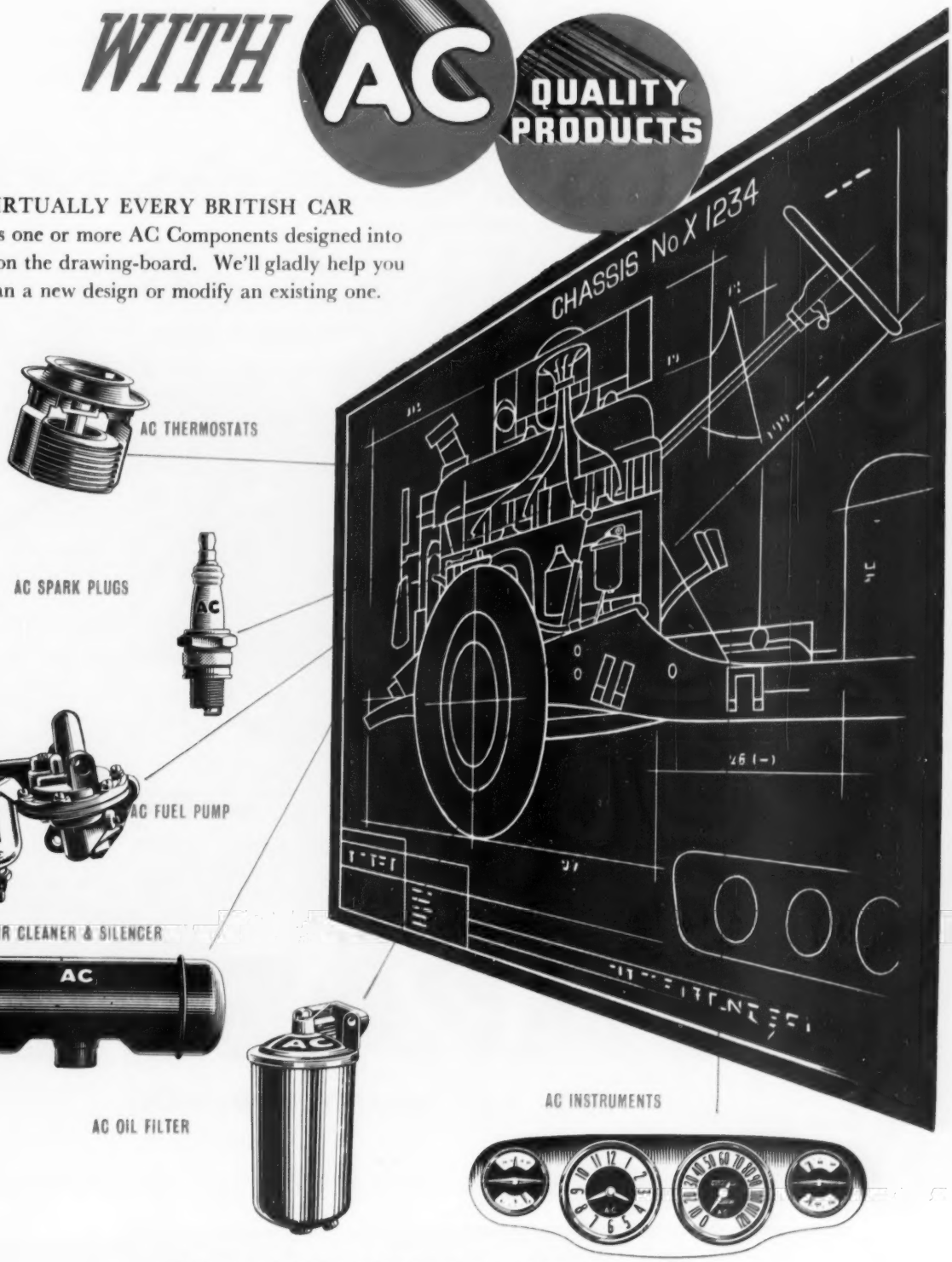
WILLIAM JESSOP & SONS LTD

Founded 1774

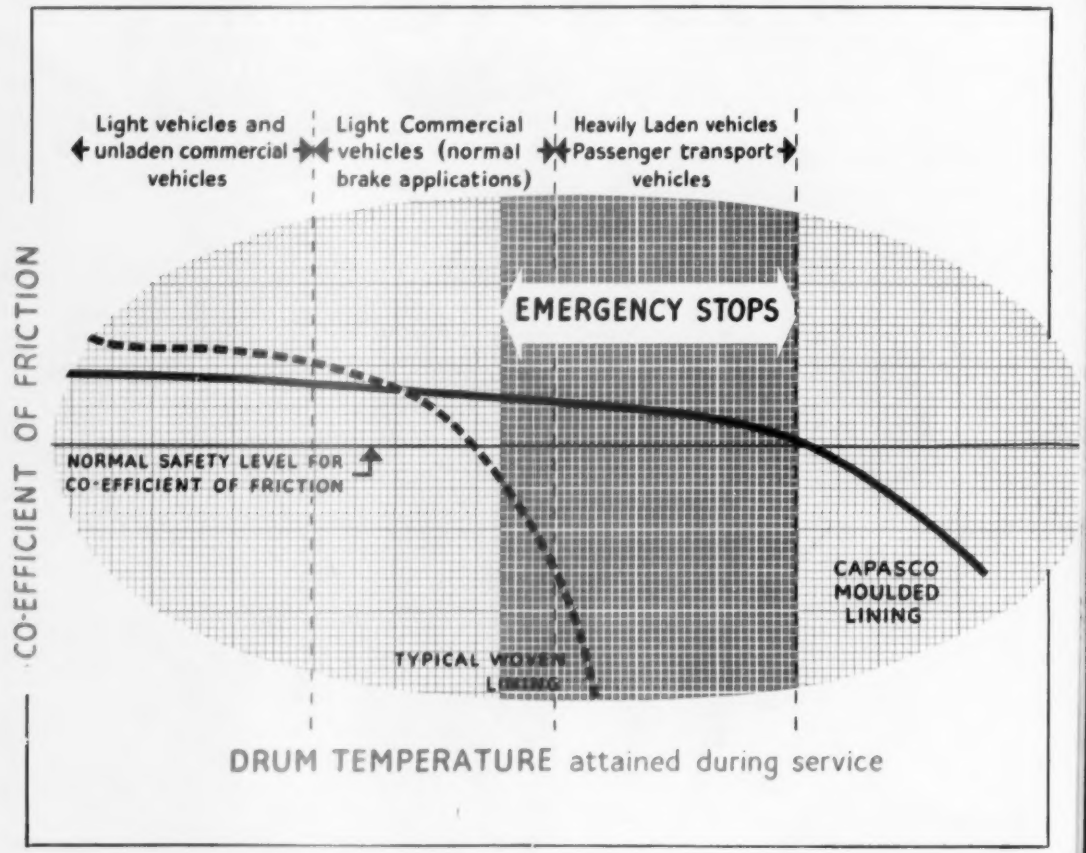
STEELMAKERS, SHEFFIELD

PLAN YOUR NEW MODELS WITH AC QUALITY PRODUCTS

VIRTUALLY EVERY BRITISH CAR
has one or more AC Components designed into it on the drawing-board. We'll gladly help you plan a new design or modify an existing one.



Write to AC-DELCO Division of General Motors Limited
Dunstable, Beds., Telephone: Dunstable 1166 or Broadgate House, Coventry, Telephone: Coventry 40491


Light vehicles and unladen commercial vehicles
Light Commercial vehicles (normal brake applications)
Heavily Laden vehicles Passenger transport vehicles
EMERGENCY STOPS
CO-EFFICIENT OF FRICTION
NORMAL SAFETY LEVEL FOR CO-EFFICIENT OF FRICTION
CAPASCO MOULDED LINING
TYPICAL WOVEN LINING
DRUM TEMPERATURE attained during service

CAPASCO

MOULDED BRAKE LININGS

The Cape Asbestos Co. Ltd., 114-116 Park Street, London, W.1. *Tel:* GROsvenor 6022

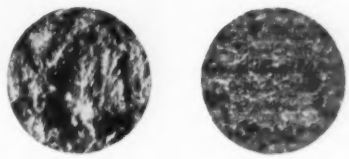
CAPASCO

moulded linings

for sustained resistance to fade

Another good reason why every CAPASCO Lining is Moulded

This graph broadly illustrates the fundamental difference in performance between moulded and woven friction materials. It can be seen that a moulded lining offers a greater margin of safety in the higher temperature range. CAPASCO is most reliable when reliability is needed most. In addition to fade resistance, CAPASCO has these other special features—Dimensional Stability—High Mechanical and Impact Strength—Rapid 'Wet' Recovery—Uniform Wear to 'Wafer' Thickness—Non-Abrasive to Brake Drums—Ease of Cleaning when Woven Linings would have to be discarded



These micro-photographs of a Woven Lining and of a CAPASCO Moulded Lining show the very striking difference in texture. It is this homogeneous structure of a Moulded Lining which is basically responsible for the excellent overall characteristics.

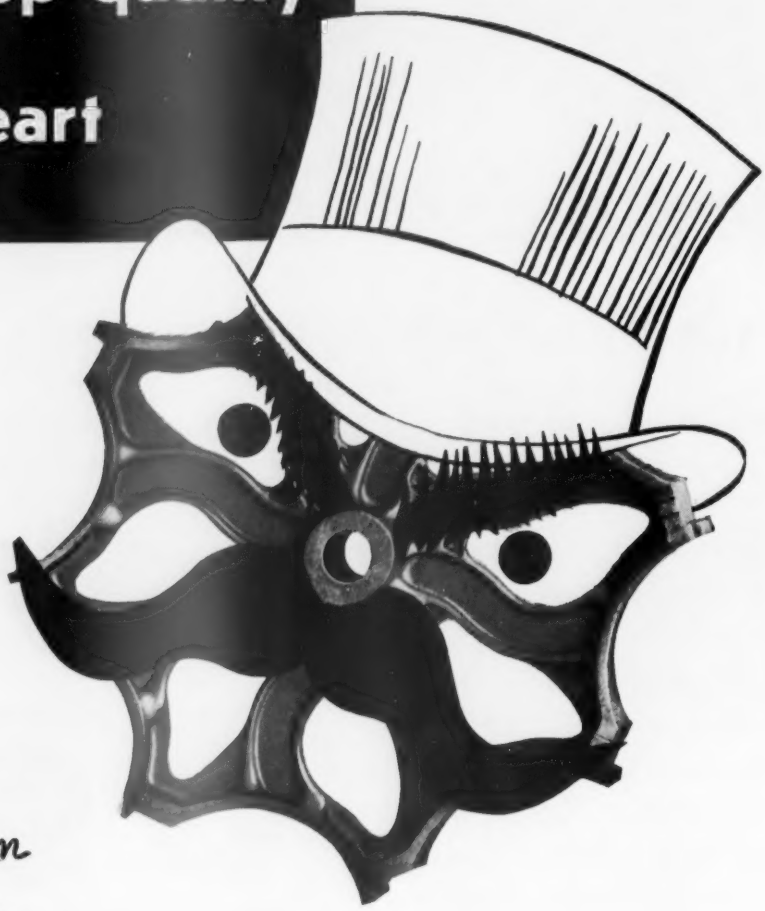
CAPASCO

MOULDED BRAKE LININGS

The Cape Asbestos Co. Ltd., 114-116 Park Street, London, W.1. *Tel:* GROsvenor 6022

TA5793

I'm a top quality blackheart



straight from Gloucester

EVERY day thousands of blackheart malleable castings flow from the mechanised line of Gloucester Foundry—every one to the customer's most exacting requirements, each with the easy machining qualities that mean lower production costs for the user.

Gloucester Foundry has an unrivalled capacity for large outputs and speedy service. It has the only elevator-type electric furnace that allows blackheart castings to be annealed in 48 hours instead of the usual 9 days. Call in Gloucester about that next casting job and save yourself time and money.

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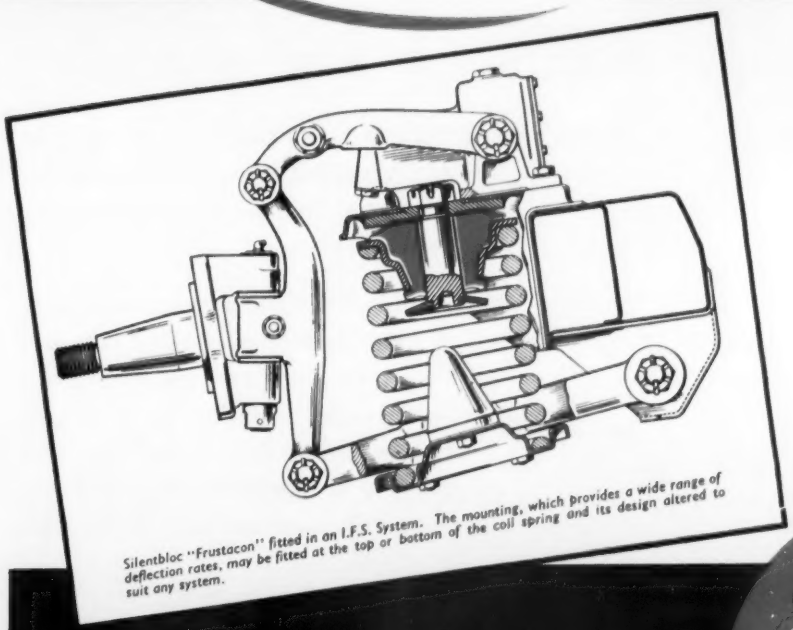
THE HOME OF
GOOD CASTINGS

Silentbloc FRUSTACON..

. . . the first scientifically designed flexible mounting fitted to the I.F.S. of a Production Car

The 'Frustacon' mounting fitted to the I.F.S. of the 1952 Rover '75' Sectioned to show rubber.



Silentbloc "Frustacon" fitted in an I.F.S. System. The mounting, which provides a wide range of deflection rates, may be fitted at the top or bottom of the coil spring and its design altered to suit any system.

To minimise the transmission of road noise to the body by the coil springs, Silentbloc Frustacon mountings have been incorporated in the I.F.S. of the 1952 Rover '75'. This notable step forward in the reduction of noise is another example of co-operation between car manufacturers and Silentbloc Ltd.

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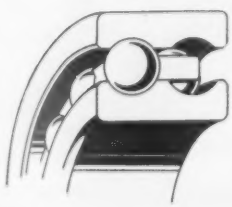

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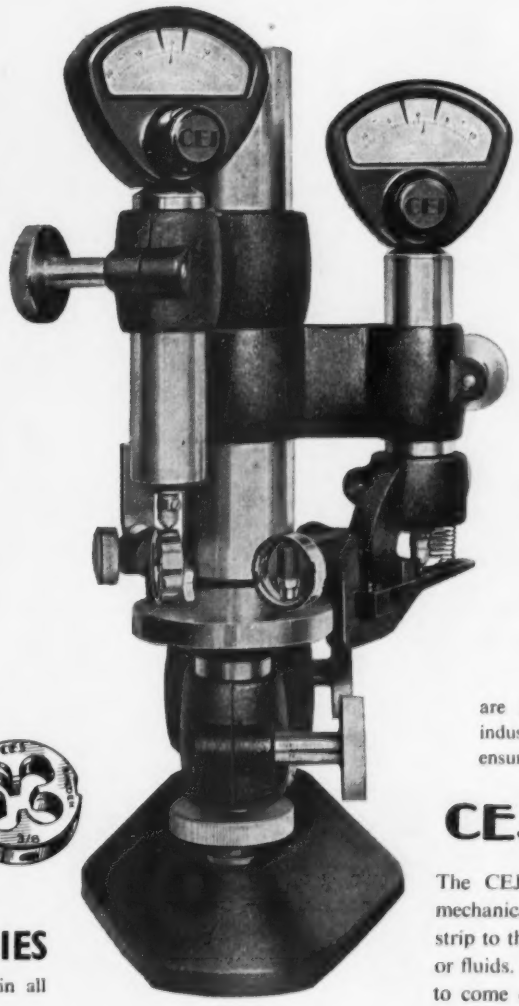

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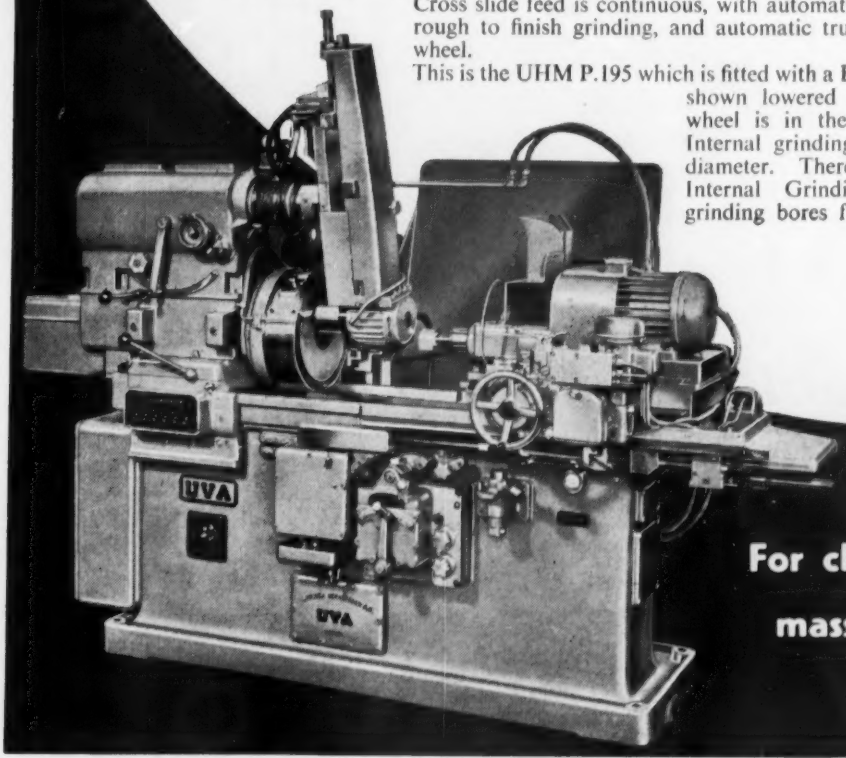
This is the UHM P.195 which is fitted with a Face Grinding Head, shown lowered so that the facing wheel is in the grinding position. Internal grinding capacity ¾″ to 8″ diameter. There are other U.V.A. Internal Grinding Machines for grinding bores from ½″ to 18″ dia.



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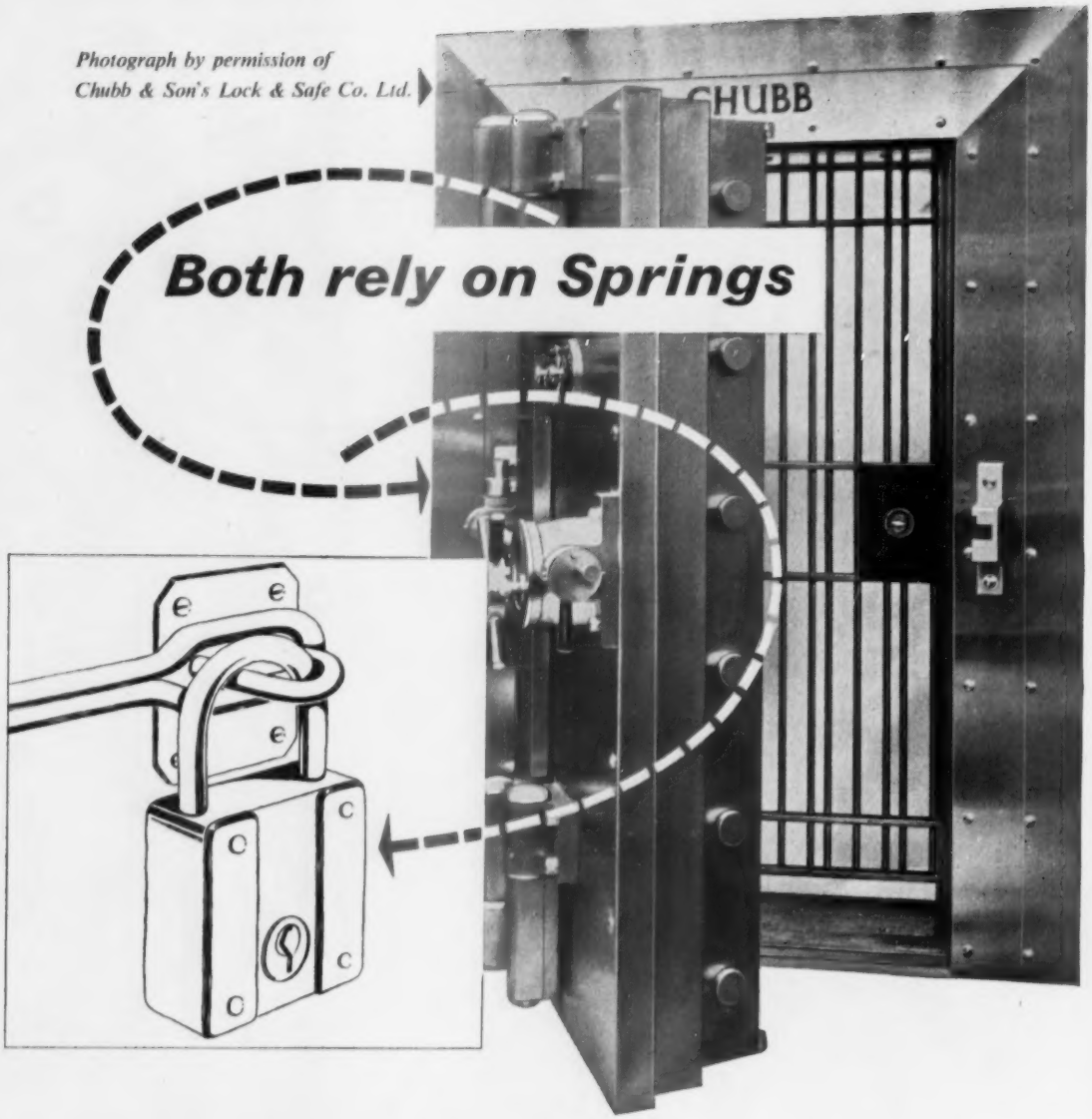

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M-W.381

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Machine and traction gear components are gas carburised in these two Birlec pit furnaces at the Park Works, Huddersfield, of David Brown & Sons. Efficient operation, low operating costs and controllable case depths characterise this range of standard equipment—borne out by the increasing number of installations throughout the engineering industry.

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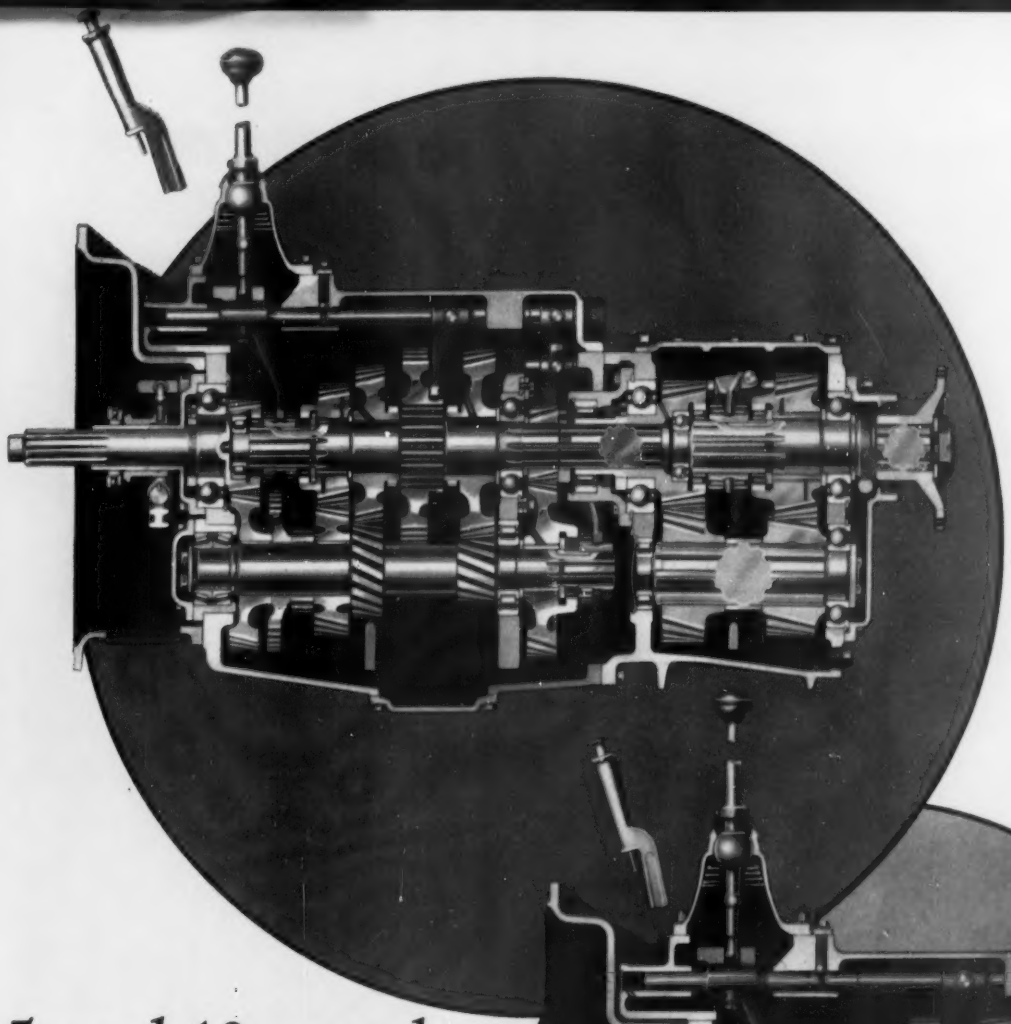

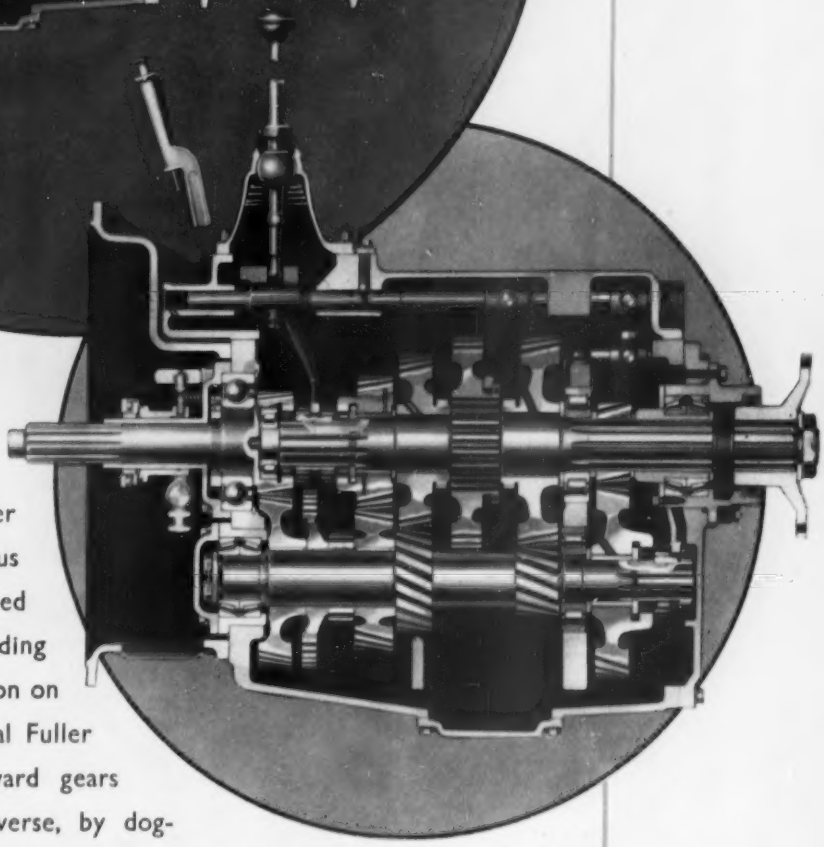

5 and 10-speed gear-boxes

The top illustration shows the Fuller 10-speed gear-box, comprising the famous five-speed box with a Fuller two-speed auxiliary box built on to it, thus providing a ten-speed box for heavy-duty operation on large trucks. This unit is to the usual Fuller standard of high-duty, with all forward gears helical, and all changes, including reverse, by dog-clutches. On both of these boxes the gears are shot-peened and crown-shaved, to avoid stress concentration.

The lower illustration shows the Fuller 5-speed gear-box. Every gear is helical and engaged by dog-clutches and to reduce shaft deflection to a minimum the mainshaft is supported on three bearings, and the layshaft kept short, making one of the highest-duty gear-boxes ever produced.

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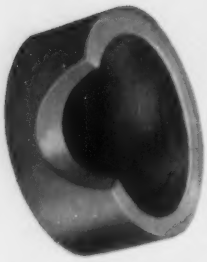
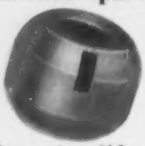
A TI COMPANY

what have these parts in common?



They are designed with a definite object in view, to provide a finished part with self lubricating properties ready for



assembly. In most cases no further attention will be necessary during the life of the equipment.

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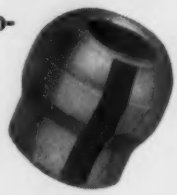

Only a few examples of

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With acknowledgements to High Duty Alloys Ltd. and the English Electric Co. Ltd.

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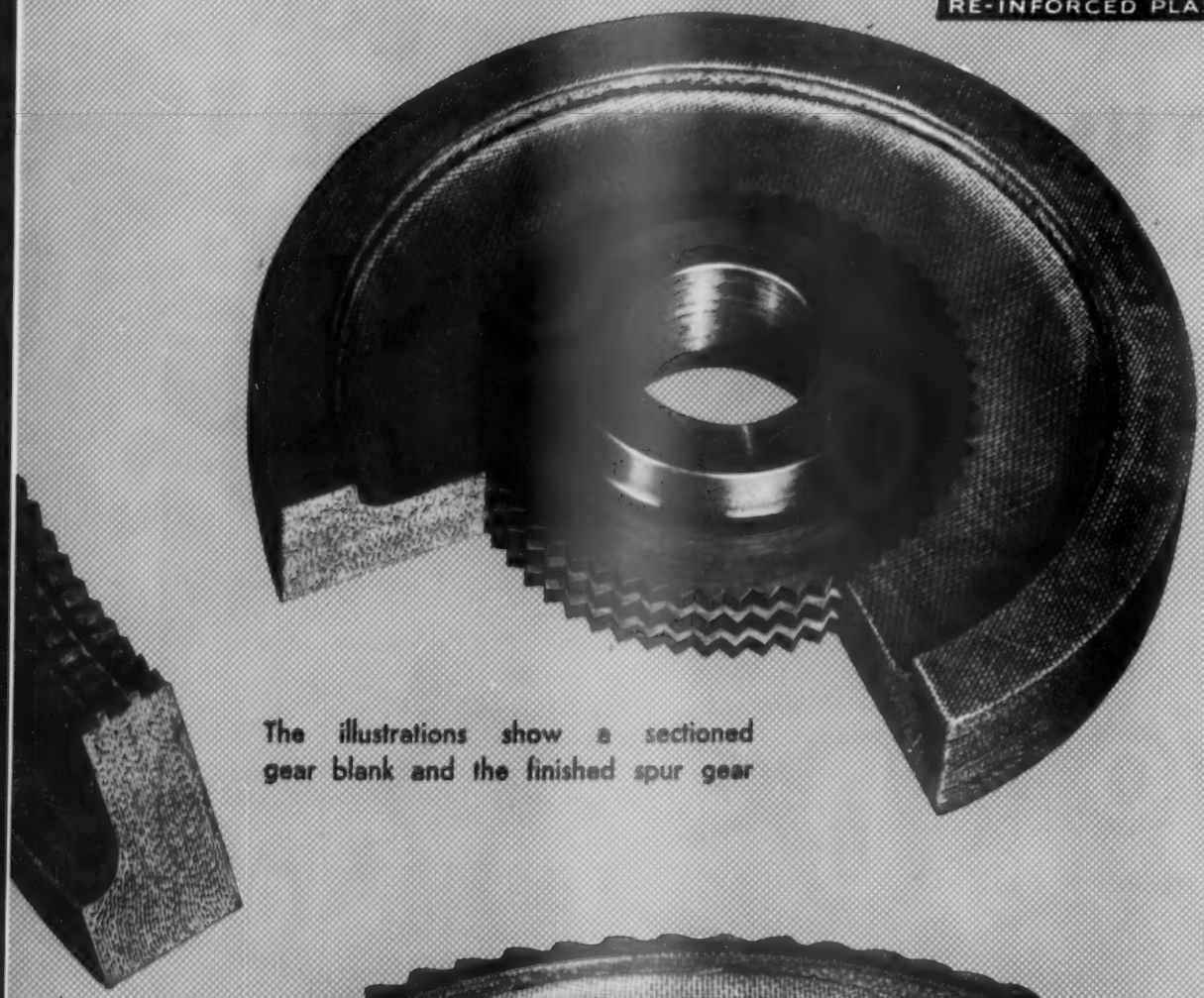
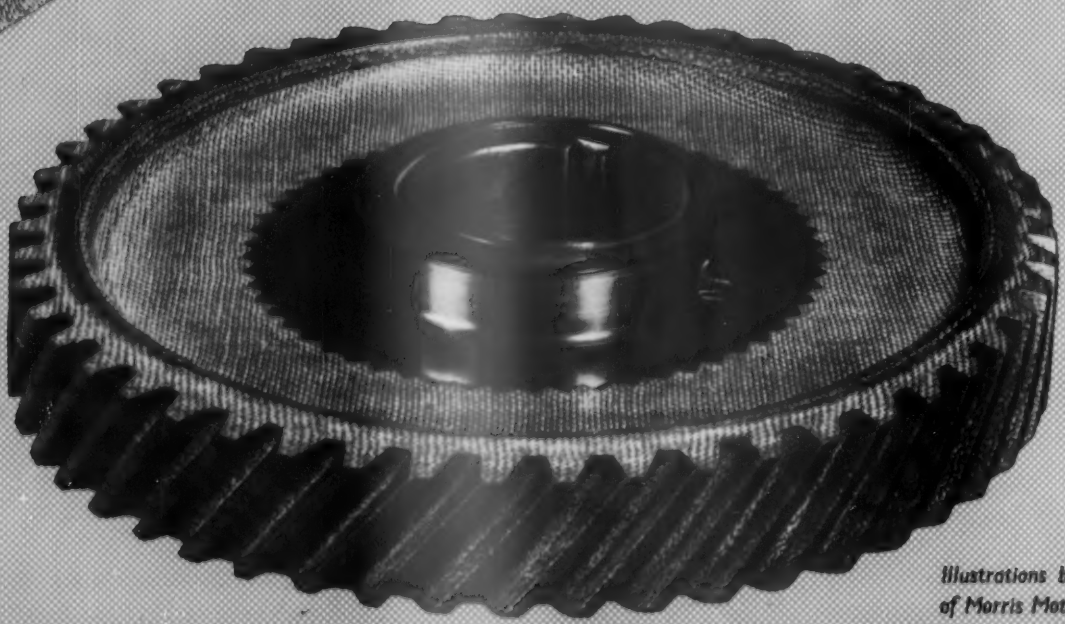


The illustrations show a sectioned gear blank and the finished spur gear



Illustrations by courtesy of Morris Motors Limited

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108 Spire Speed Nuts are in use throughout the new Armstrong-Siddeley Sapphire. These nuts make faster, easier assembly — and that means a saving of labour and materials. They do away with the need for expensive welding, riveting, clinching and the use of threaded inserts. And apart from functioning better, they save between 50% and 75% on weight of materials used. Altogether there are over 200 different types of Spire Speed Nuts. On the Sapphire, however, these are the main types and some of the functions of the parts that are used:—SNU Nuts for securing the bonnet hinge to the bulkhead; SNJ Nuts for securing the front wings to the front pillar; SCL Clips for securing the nameplate; SFP Fixes for securing the radio grille to the dashboard; and SM/F Grips for securing the front seats to the floor. All these jobs, and many others besides, can be done faster, cheaper and better by Spire.

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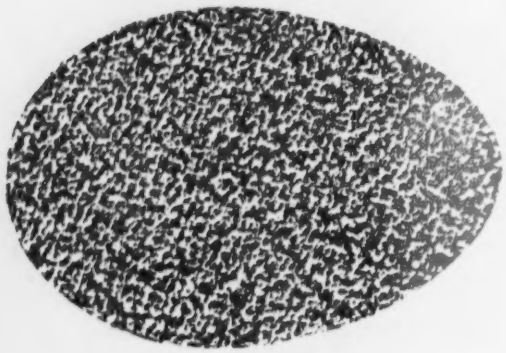
The Hen...



or the Egg ?

Which came first? It's quite a query, since each connotes the existence of the other. And so it is with packaging: the processes from first to last are interdependent. Inner carton or outer case. Wax-wrapping or labelling. Market presentation or tropicalization. REMPLOY P.I.P. SERVICE *includes them all*. It is ready to lay a preliminary design before you or to hatch out a complete packaging plan worth clucking over!

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Including Ductile Iron

are casting Alloy Steels and High Duty Irons

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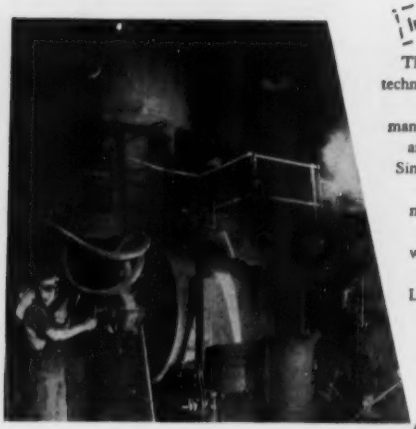

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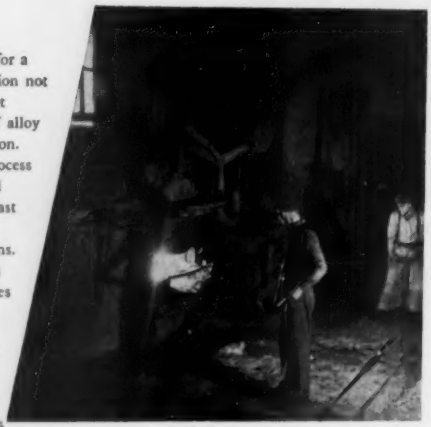

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The transformer is rated for automatic welding at the top setting.

Special plants recommended for the welding of nickel and copper alloys are available for inert gas welding with D.C.

600 amp. Argon Arc welding plant with automatic controls.

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Photograph by courtesy of Ford Motor Company Ltd., Dagenham, shows the multi-drilling of tractor cylinder blocks.

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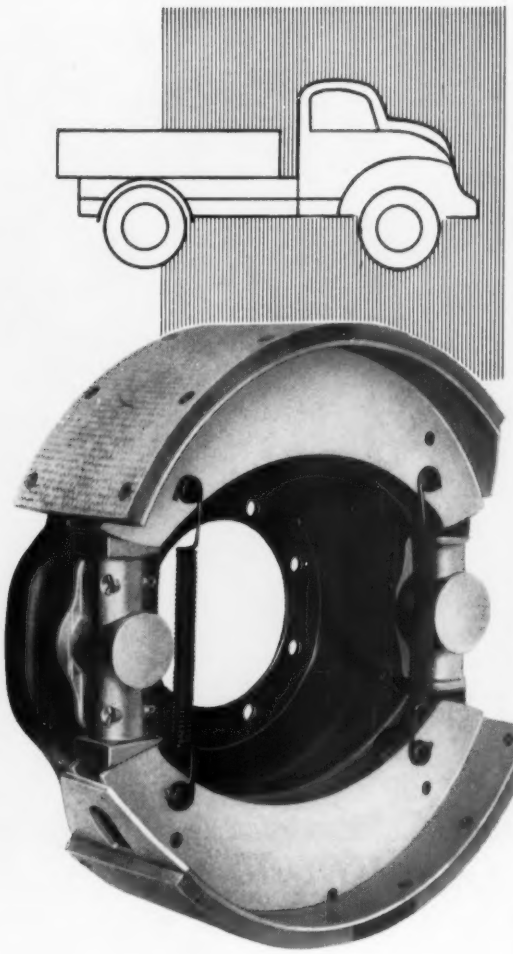

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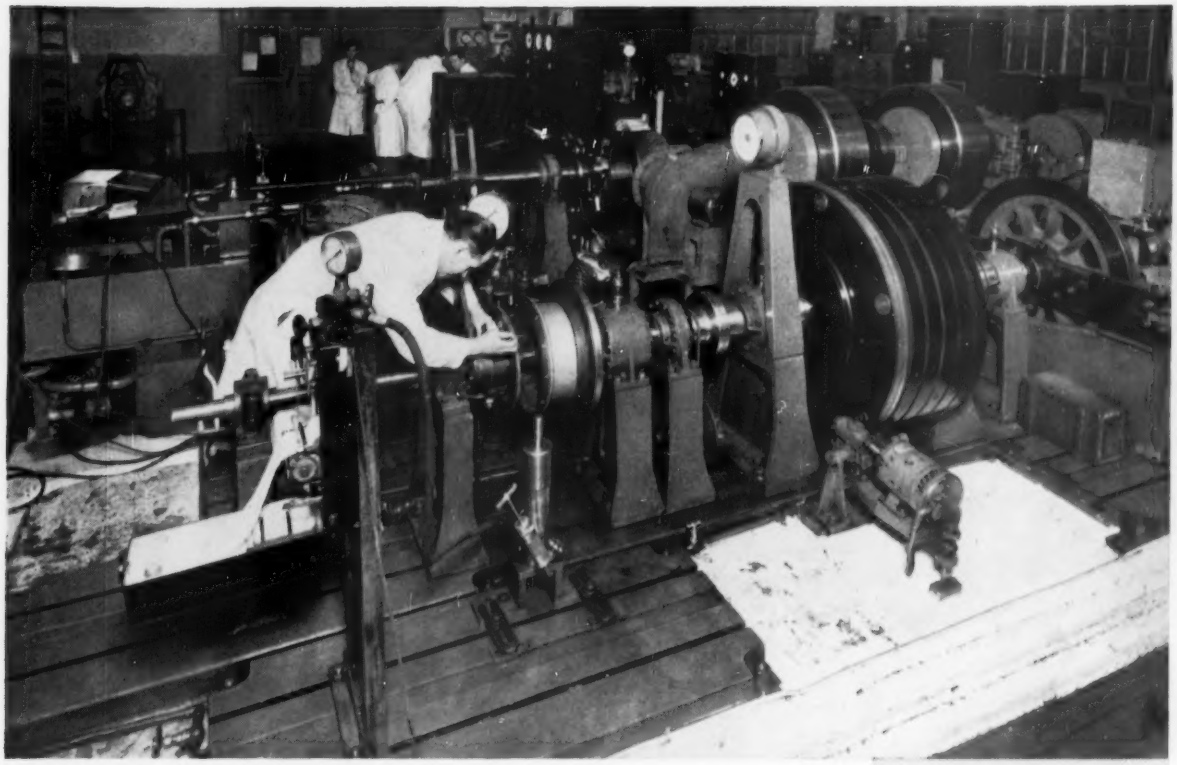

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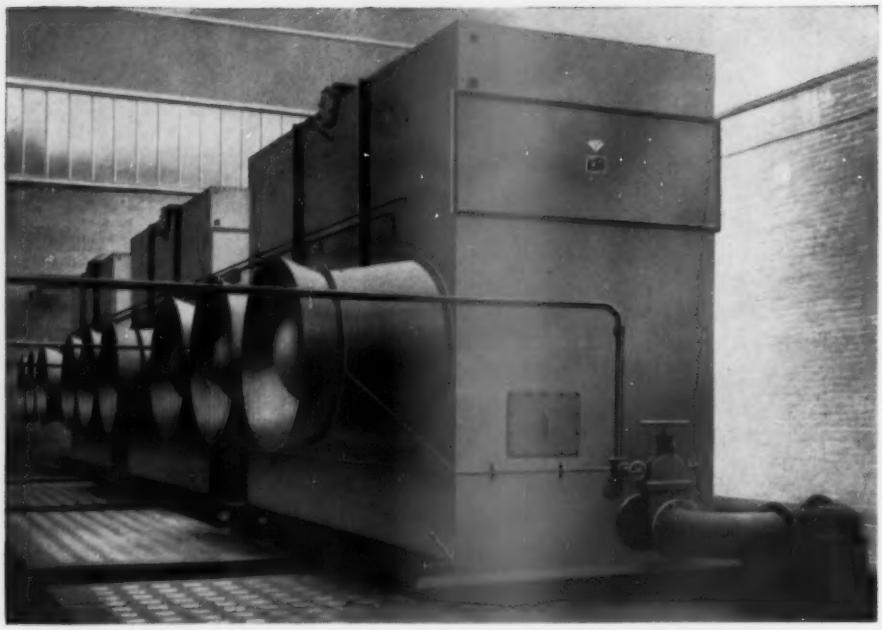

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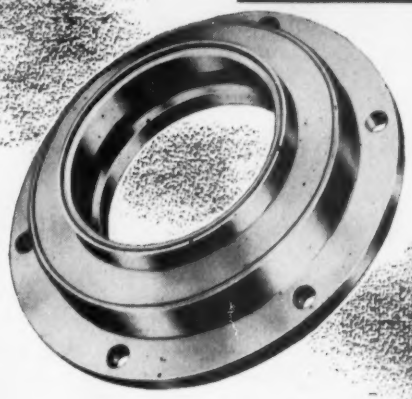

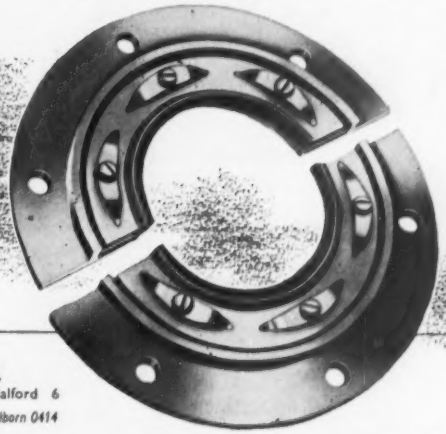

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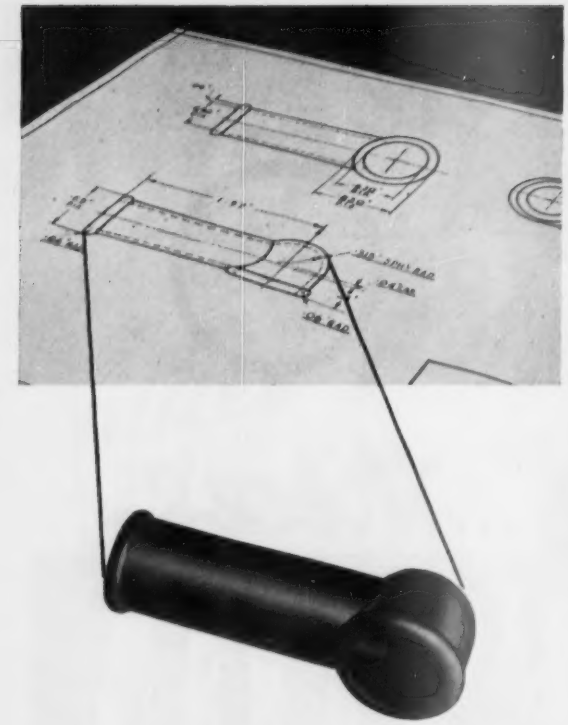

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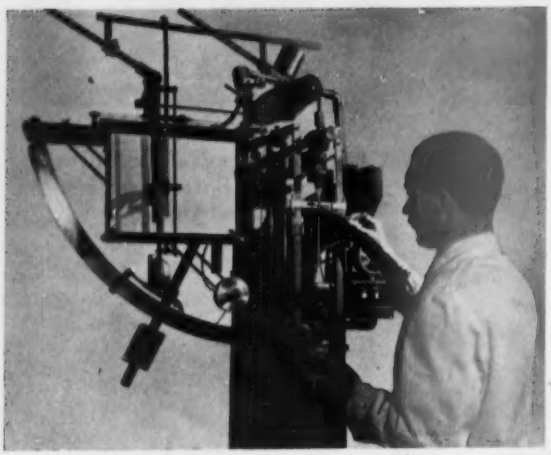

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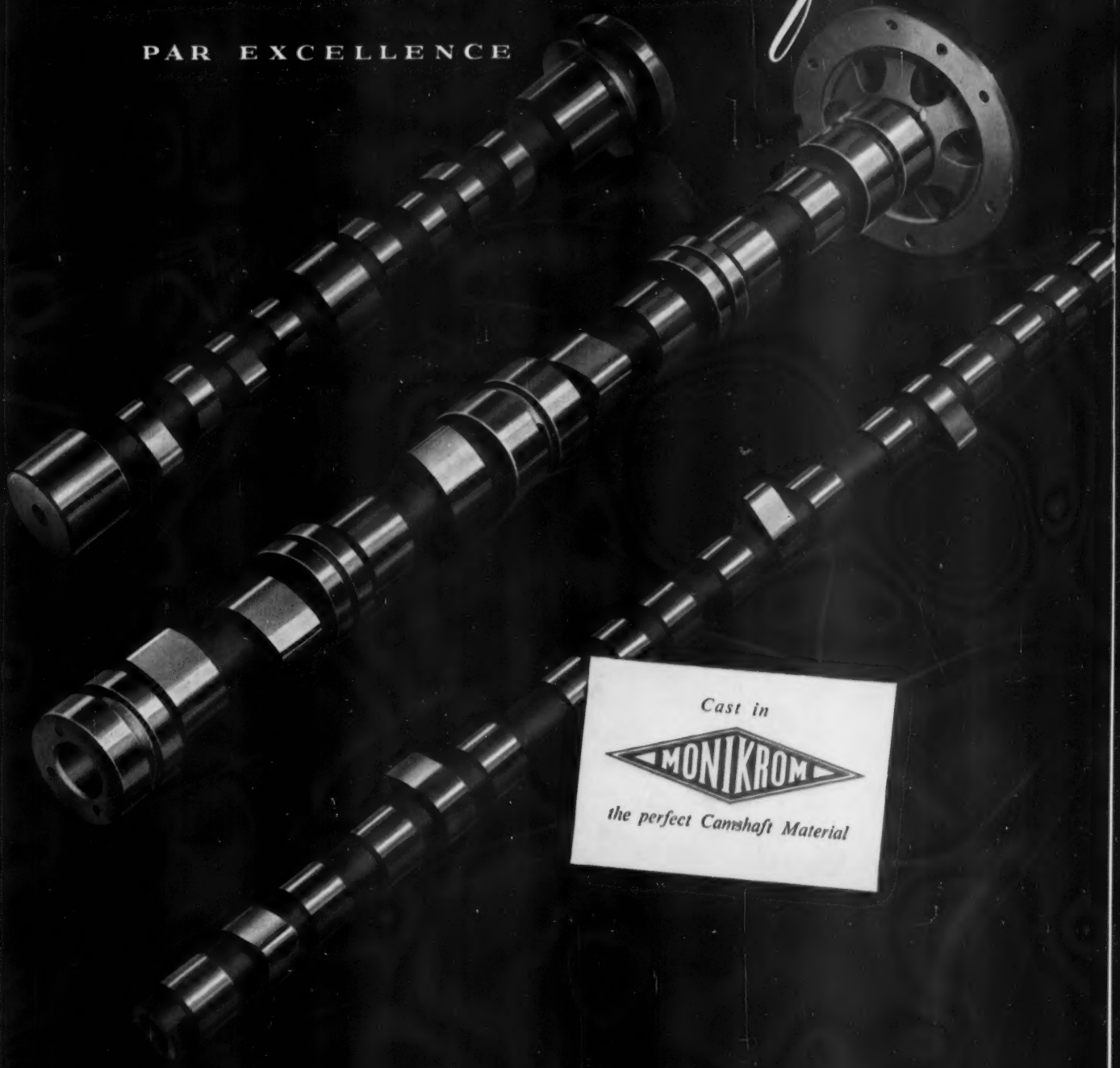

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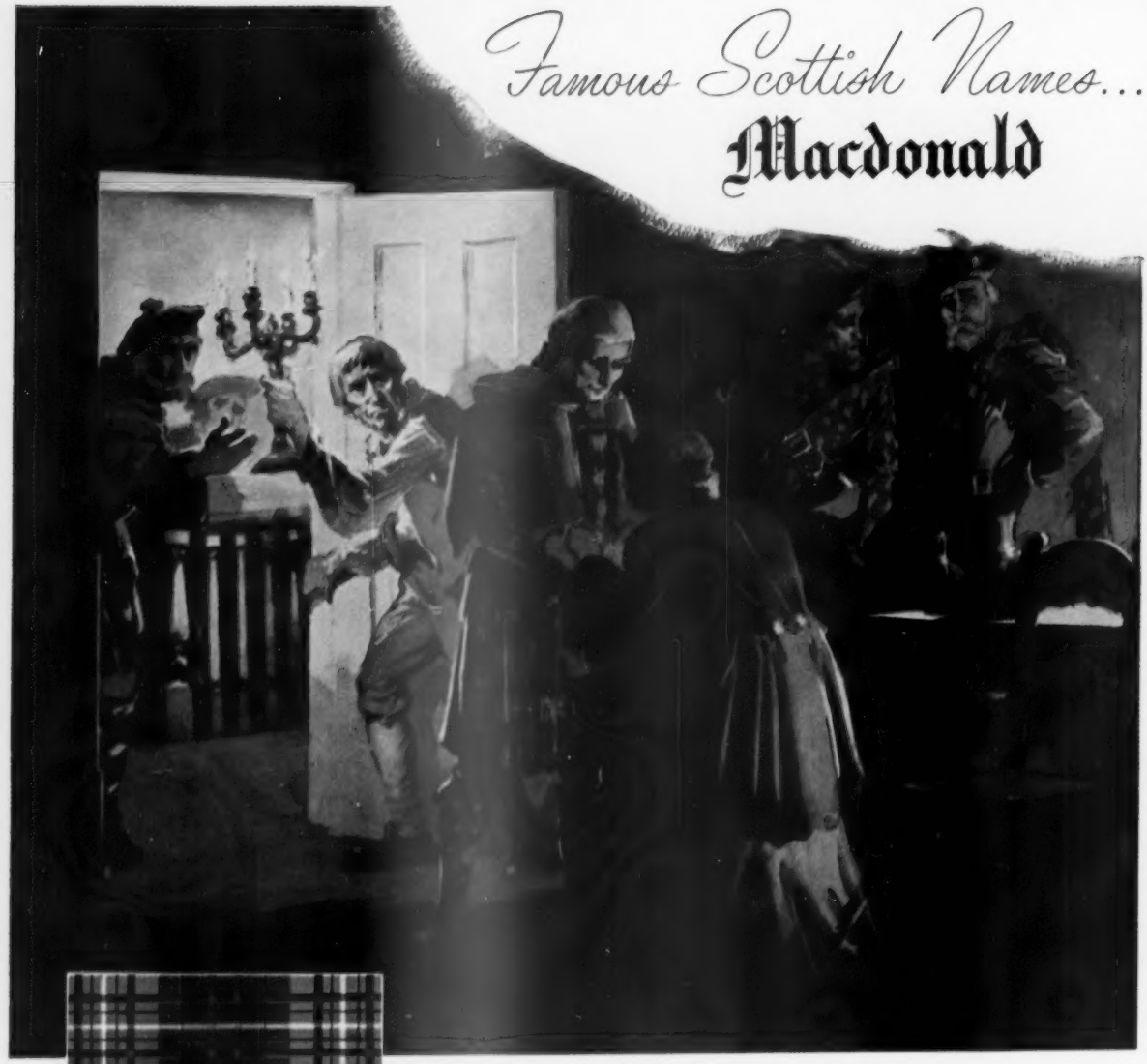

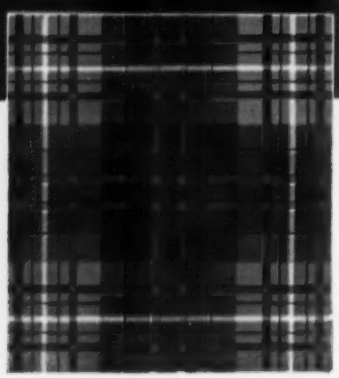

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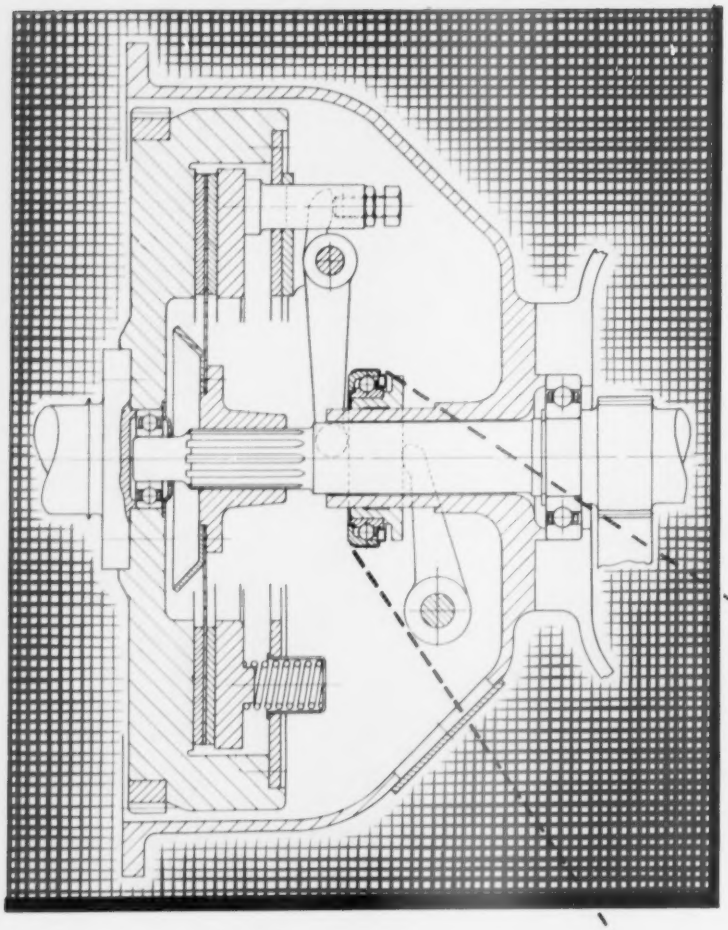

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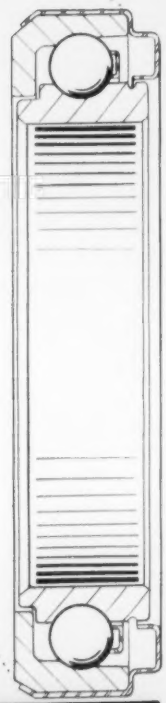

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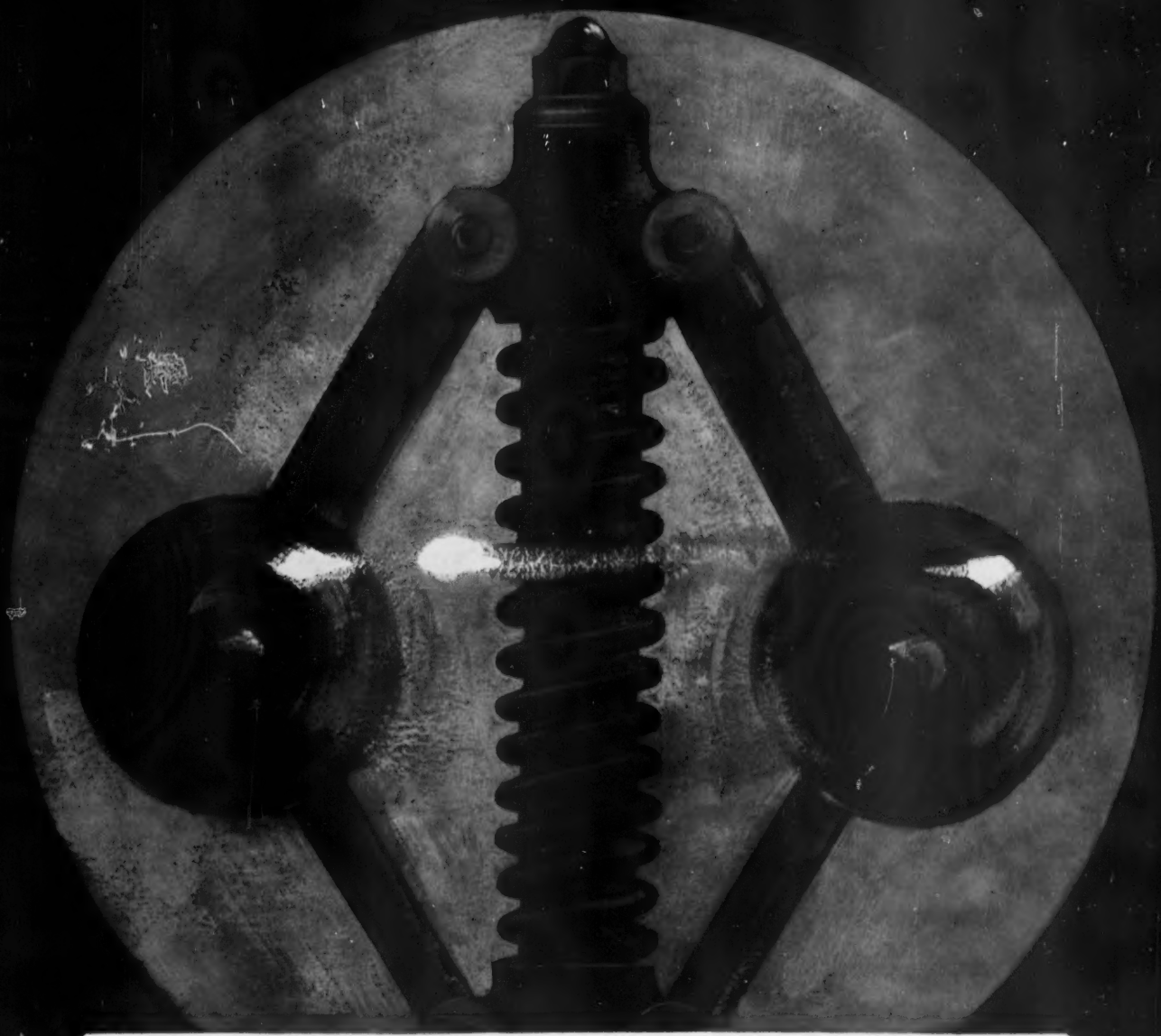

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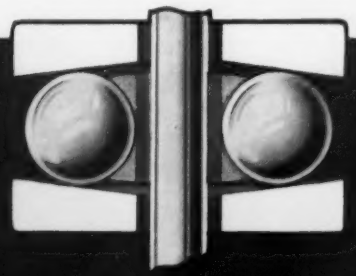

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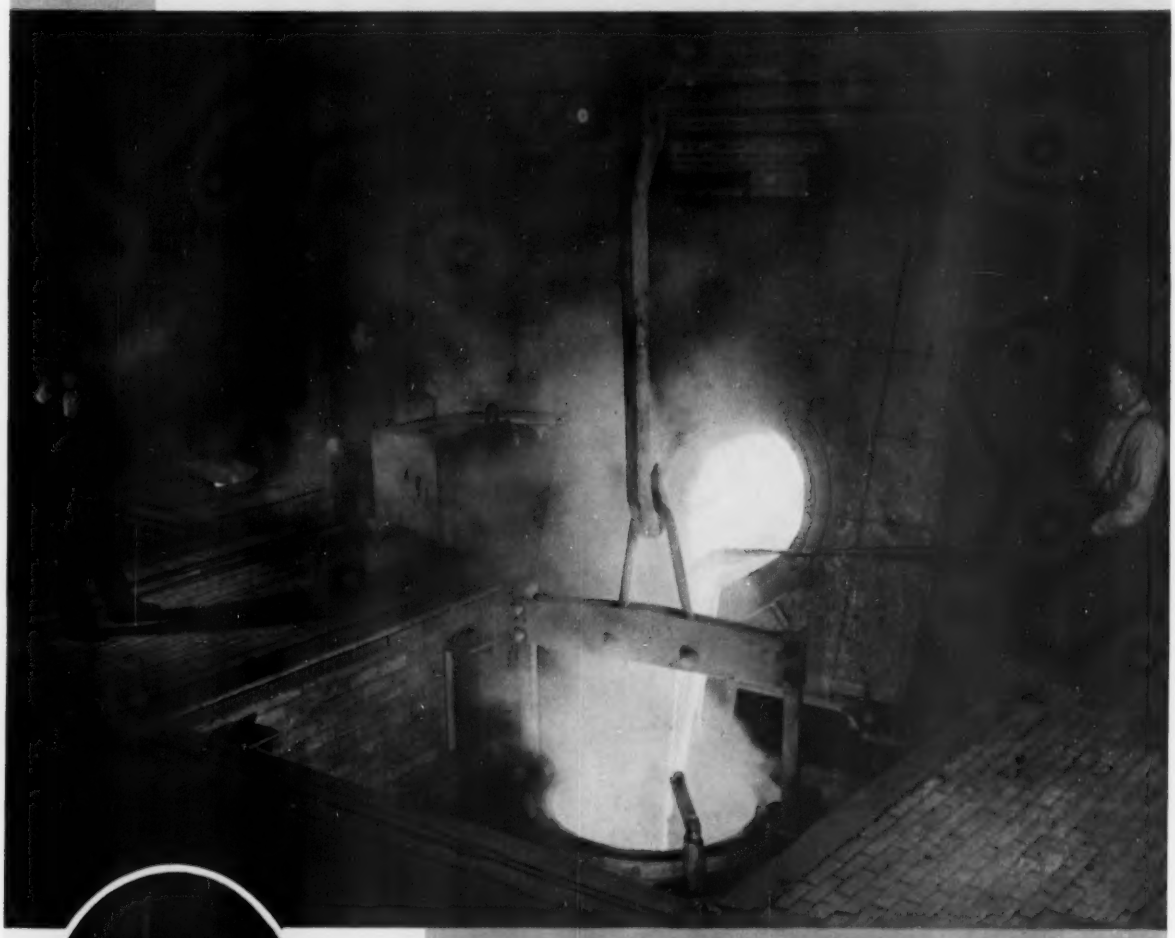

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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

Editor: J. B. DUNCAN

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Publishers: ILIFFE & SONS LTD., DORSET HOUSE, STAMFORD STREET, LONDON, S.E.1.
Telegrams: Sliderule, Sedist London **Telephone: Waterloo 3333 (60 lines)**

COVENTRY:	BIRMINGHAM, 2:	MANCHESTER, 3:	GLASGOW, C.2:
8-10, CORPORATION ST.	KING EDWARD HOUSE, NEW ST.	260, DEANSGATE	26B, RENFIELD ST.
Telegrams: Autocar, Coventry	*Telegrams:* Autopress, Birmingham	*Telegrams:* Iliffe, Manchester	*Telegrams:* Iliffe, Glasgow
Telephone: Coventry 5210	*Telephone:* MIDland 7191-7	*Telephone:* Blackfriars 4412	*Telephone:* Central 1265-1266 (2 lines)

PUBLISHED MONTHLY—SECOND WEDNESDAY. ESTABLISHED 1910

Annual Subscription: Home and Overseas £2 11s. 6d., including the Special Number; Canada $7.50; U.S.A. $8.00

VOL. 44 NO. 7 JULY, 1954 PRICE 3s. 6d.

An Honour Well-merited

ALL automobile engineers will join in extending heartiest congratulations to Mr. William Lyons, Chairman and Managing Director of Jaguar Cars, Ltd., on his appointment as a Royal Designer for Industry. The very restricted membership of this Faculty, it must not at any time exceed 40, includes outstanding personalities from many fields, among them, Sir Geoffrey de Havilland and Sir Hugh Casson. Mr. Lyons is a worthy addition to their number.

While the honour is a personal one, it also reflects credit upon the recipient's colleagues, since there is not the slightest doubt that the outstanding success achieved by Jaguar cars is the result of splendid team work under the captaincy of Mr. Lyons. The whole of the automobile industry will welcome the honour paid to one of its members for two reasons. First, because it is so well merited; and secondly, because it shows recognition that artistry, as well as science, can be employed in the development of the modern automobile. The Council of the Royal Society of Arts are to be congratulated on making an appointment that will bring added prestige to the Faculty of Royal Designers for the Industry.

Vehicle Lighting

PRESENT-DAY conditions on the roads at night create problems of great urgency. The general increase in the cruising speeds of modern cars and the higher density of traffic on British roads are factors that have made the problems much more serious than they were before the war; and in future we can expect automobiles capable of even higher cruising speeds and greater traffic density. Although lighting systems are still far from ideal, lighting engineers have effected considerable improvements over pre-war practice by the application of developments in lamp-filament and projector design and increased optical knowledge. Nevertheless, much remains to be done to make night driving safe and pleasurable.

In dealing with automobiles, the lighting engineer has to effect a compromise between what are virtually two incompatibles; that is, he must seek to eliminate dazzle and at the same time provide sufficient illumination for safe driving at relatively high speeds. With present knowledge and techniques, either problem alone can be solved, but when they are dealt with in conjunction some compromise is inevitable.

Since discomfort caused by glare from oncoming cars is a physical fact, while safety is an intangible, most road users consider that the elimination of dazzle is the prime desideratum. This is a mistake; adequate forward vision is the first essential. Even when meeting oncoming traffic, safety demands that the beam must give adequate illumination over a certain minimum length of road ahead, before the reduction of dazzle is considered.

British practice, which is closely akin to American, differs considerably from Continental practice as to the type of distribution of the so-called meeting beam to give adequate visibility with reasonable freedom from dazzle. The usual British present-day solution to this problem is severely to limit the candle-power in the direction of the oncoming vehicle by deflecting the beams of the two headlamps to the nearside, in such a manner that the driver is given a reasonable range of vision along the nearside kerb, together with a pool of light immediately in front of the vehicle to allow the car to be positioned accurately on the road.

This type of lighting system necessarily restricts the range of illumination when the lamps are dipped, but the restrictions are those of geometry and not those of lighting engineering. On a modern vehicle, the headlamps are usually mounted about 30 in above the ground, and because of vertical movement due to the softness of the suspension and undulations of the road, a long range meeting beam necessarily causes great discomfort to, and can imperil, the oncoming driver.

Under almost all conditions an oncoming driver's lamps must represent a glare source that must cause some discomfort, but if they are double-dipping, they do disclose the offside kerb and fill the road surface between the meeting vehicles. It is only within recent years, that the practice in this country, as in the rest of the world, has been to use both headlamps when meeting an approaching vehicle. Double-dipping headlamps have been, and still are, the subject of controversy, but they do provide the best available compromise for giving the optimum combination of visibility and freedom from dazzle. Recent developments in headlamp optics have made possible the introduction of a standard meeting beam suitable for all vehicles, that has been adopted by manufacturers for most British vehicles intended for both the Home and Commonwealth markets. A standard meeting beam will add to the comfort and safety of night driving.

In many ways, the side lamps, rear lamps and stop lamps, or as they are known in international parlance, the indicating

lights, are of more importance to road safety than the headlamps. Recent developments have greatly improved these accessories. The use of durable moulded plastics has made it possible to devise an optical system of an accuracy that is impossible with ordinary glass, while research carried out in all parts of the world has set standards of performance for tail lamps.

Before the war, the average candle-power given by a tail lamp was something in the order of 1/50th of a candle. In 1947, the American S.A.E. specification called for 1/10th candle, a requirement that both British and American industry had difficulty in meeting. Owing to improvements in manufacturing techniques, plastics moulding, colour transmission control and more accurate filament manufacture, the 1953 draft specification of the I.S.O. calls for 1½ candle-power. It can be met with much greater ease than could the far lower 1947 specification with the techniques then current. Even so, it would appear that the factor of safety is none too great for adverse conditions of rain and glare.

Military Vehicle Maintenance

A MAJOR worry for the Army Council today is that about one-half of the total army strength is engaged on support duties, so that the "tail," to use the recognized army term, is now as long as the dog. Because of the complexity of the equipment used by a modern army, the ratio of supporting troops to fighting troops must be relatively high, but it is probable that the present ratio might be reduced without any loss of efficiency. There is no doubt that the automobile industry could, if asked, supply the "know-how" necessary for using support troops with greater efficiency and therefore in lesser numbers.

Among other duties, R.E.M.E. is responsible for the major servicing and maintenance of all motor vehicles. The magnitude of this problem alone may be judged from the fact that the reserve stock of vehicles is larger than was the total fleet, at its numerical peak, of the British Road Haulage Executive. In every day of a five-day working week there is an average turnover of 1,000 vehicles in and out of depots. During 1953, 95,000 vehicles were serviced.

The maximum possible simplification and standardization of equipment is obviously the first step that should be taken. This should be followed by the development of a small number of home-based, servicing depots for all vehicles, except perhaps the heaviest armoured fighting types. Because of the volume of work that would pass through these depots, it would be economically practicable to reduce the amount of labour required by installing conveyor lines and other mechanical handling devices. It should also be possible to break down into its simplest elements each servicing operation so that the maximum possible use can be made of unskilled and semi-skilled labour. This would effect a considerable economy in training after mobilization as well as time.

It is possible that certain objections might be raised to the use of these methods. One is that if a depot were damaged by enemy action, it would be difficult to get going again. This is a matter of balancing alternatives; whether it is better to have a relatively small and somewhat vulnerable "tail" or a much larger and less vulnerable and much less efficient "tail." In any case, the capacity of the maintenance services must be related to the flow of original equipment, and efficient production methods are as essential for the maintenance as they are for the supply of equipment to a mechanized army.

The automobile industry long ago found that the best, if not the only, way to make use of unskilled labour on complex machines, is to break down service operations into their simplest elements. Economic factors have forced this solution on industry, but unfortunately such factors do not have the same compelling force in the Services. Nevertheless, even in peacetime, and much more so in wartime, the army must make-do with large numbers of unskilled personnel, whom it is impossible to turn into skilled men in the short time that can be allotted to training. The logical procedure is to follow automobile practice in planning the work to suit the available labour. This would entail the employment of a specialist planning staff to study the work in detail, but should lead to an appreciable increase in output with labour that required little training.

Undoubtedly, much has been done in mechanizing army stores and servicing units, but much remains to be done if the best possible use is to be made of the labour available. The experience of engineers throughout the automobile industry could be of great assistance.

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THE STANDARD EIGHT

Part II. A Chassis of Unitary Construction Designed for Ease of Repair

THERE is no resemblance between the new Standard Eight and any other model previously produced by Standard Motor Co. Ltd. This is not surprising, for considerable technical progress has been made since the last 8 h.p. Standard car was in production, and the aim in designing the new model was at embodying many features that have been well proved during the intervening years, as well as further improvements resulting from more recent experience.

It is doubtful if any other vehicle of unitary construction and with full width bodywork of modern form incorporates so many readily detachable panels and main structural members. This would appear to be one answer to the criticism, often levelled against unitary construction, that repair of damage due to collisions is a major operation and in many instances impossible. In comparison with pre-war models, deterioration due to corrosion should be considerably reduced, since the bodywork is now Bonderized before painting. Not only does Bonderizing itself protect the steel against corrosion but it also produces a surface to which the paint film subsequently applied will key firmly.

Primarily, this vehicle has been designed to meet the demand for an economy car; for those wanting a less austere vehicle, a 10 h.p. model has been developed. To design an acceptable economy car has not been an easy task, for while customers require a car at a low price, they also expect at least some of the amenities of the more expensive class of vehicle. In great measure, the Standard Eight reconciles these contradictory requirements. The car has plenty of room inside for four passengers. A radio and heater can be fitted as optional extras, and there is ample space for luggage.

To keep the price to a minimum, a boot lid is not provided, and access to the luggage compartment is gained by pulling forward the rear seat squab, which is hinged at its base. When the two rear seats are unoccupied they can be folded up against the front seat squabs to make room for additional luggage. Horizontally sliding windows are incorporated to alleviate the need for the actuating mechanism that would have been required to raise and lower vertically sliding types, and to enable the trim to be fitted directly on to the inner face of the exterior panel. Triplex Safety Glass is used for all the lights, including the windscreen. Another economy that has been effected is the reduction to a minimum of chromium plated ornamentation. For instance, there is no chromium plating round the radiator duct.

SPECIFICATION

CLUTCH: Borg and Beck single dry plate, 6½ in diameter, hydraulically actuated.

GEARBOX: Four forward speeds and one reverse, with synchromesh on top, third and second. Ratios: top 1:1, third 1·454:1, second 2·46:1, first and reverse 4·271:1.

REAR AXLE: Semi-floating hypoid pinion, with banjo casing. Ratio 4·875:1.

REAR SUSPENSION: Semi-elliptic leaf springs. Armstrong, lever type DAS8R shock absorbers.

FRONT SUSPENSION: Independent, double transverse wishbone links and coil springs. Girling telescopic DAS4 shock absorbers.

STEERING: Burman worm and nut. Ratio: 14:1 straight ahead, 12·7:1 full lock. 2¼ turns from lock to lock. Turning circle 32 ft.

BRAKES: Front: Girling hydraulic, two leading shoe type. Rear: Girling hydraulic, one leading and one trailing shoe type. Drum diameter 7 in. Shoe width 1¼ in.
Total friction lining area 68 in², front and rear areas each being 34 in².

TYRES: 5·20×13·00. Pressure, fully laden, 22 lb/in² at the front and 24 lb/in² rear. Disc type wheels.

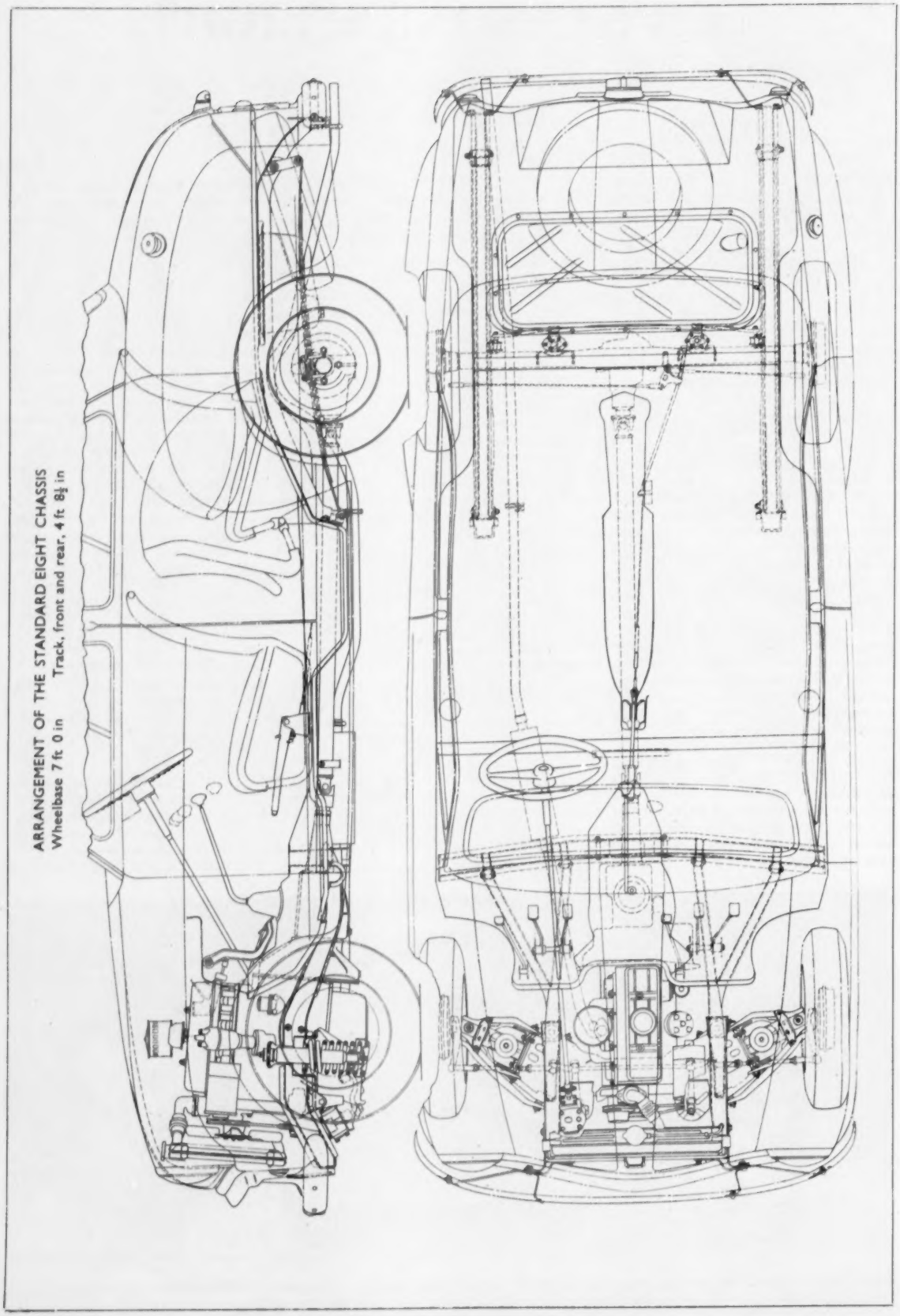
DIMENSIONS: Wheelbase 7 ft 0 in. Track 4 ft ½ in at front and at rear. Ground clearance 7 in. Overall length 11 ft 10 in. Overall width 4 ft 10 in. Overall height unladen 5 ft 0 in. Frontal area laden 19·5 ft². Dry weight 1,475 lb with tools but without extras. Front/rear weight distribution, kerb, 58/42 per cent, 50½/49½, with 4 occupants. Engine dry weight 248 lb. Gearbox and clutch weight 39½ lb.

Clutch and gearbox

A Borg and Beck single dry plate clutch is employed. The friction lining is 6½ in diameter and its area is 31·7 in². Six pressure springs are fitted; the total load on all six springs, when they are assembled in the clutch, is 220 lb. A carbon type thrust washer is used and, since the clutch is hydraulically actuated by the Girling control, a compensating device or engine tie is unnecessary.

The bellhousing is integral with the

Small cars are undoubtedly the most difficult subject that the stylist has to deal with. Despite this, and the fact that the Standard Eight has been designed with a view to keeping the prime cost to an absolute minimum, its general appearance is of a smart and well proportioned car



ARRANGEMENT OF THE STANDARD EIGHT CHASSIS
Wheelbase 7 ft. 0 in Track, front and rear, 4 ft. 8½ in

D.T.D.424 die-cast aluminium gearbox. Only three bolted-on covers are fitted to the box. One retains the front ball bearing, the second is formed by the rear extension and the third is the top cover, which houses the selector rods and striker. The gear ratios are: top 1:1, third 1·454:1, second 2·46:1, first and reverse 4·271:1.

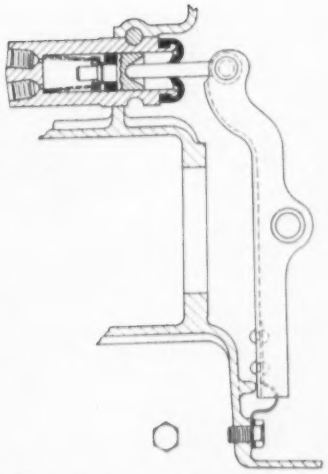
An En 18C primary shaft is employed. Its front end is turned down to 0·494-0·492 in diameter to spigot into the Oilite bush in the tail end of the crankshaft. Behind the spigot is a $\frac{7}{8}$ in diameter splined portion, on which the clutch plate is carried. To the rear of this, the shaft is reduced to 0·85 in diameter to clear the clutch thrust bearing. It is then increased to $\frac{11}{16}$ in diameter. A scroll type oil return machined in the front cover surrounds the periphery of this part of the shaft.

The front cover is bolted to the front wall of the gearbox and is pulled against the outer race of the ball bearing that carries the shaft. A snap ring in a groove round the outer periphery of the race bears against the front face of the gearbox casing and axially locates the primary shaft assembly. The inner race is carried on a 1 in diameter portion of the shaft. Its rear face is located against the oil baffle interposed between it and the primary gear. In front of the ball race is an En 1A ring with an annular groove in its inner periphery. The ring and the inner race of the ball bearing are retained by a circlip in a groove round the shaft. Surrounding this ring is another one, pressed into the front cover. This second ring also has an annular groove round its inner periphery, but an oil return scroll is machined on the lands in the bore on each side of the groove. The scrolls are so arranged as to keep the oil in the groove.

The function of this assembly round the primary shaft is to pass oil from a vertical duct in the front cover and deliver it into a counterbore drilled from the rear end of the shaft. Lubricant passes through a radial hole in the outer ring into the annular groove in its bore, and thence through holes into the annular groove round the inner ring, whence it goes through another radial hole into the counterbore. The way in which oil is passed along the layshaft and into the vertical duct in the front cover will be described later, in the section dealing with the layshaft assembly.

A $\frac{5}{32}$ in wide groove is machined round the shaft immediately behind the primary gear. The teeth and inner cone of the synchronizer are machined to the rear of the groove, which is incorporated merely to provide clearance for shaping and shaving the gear. A Clevite 10 bush that carries the front end of the mainshaft is fitted in the counterbore. This bush is 1·197-1·177 in long by 0·6255-0·6245 in outside diameter by 0·5020-0·5010 in inside diameter. The diametral clearance is 0·0010-0·0025 in.

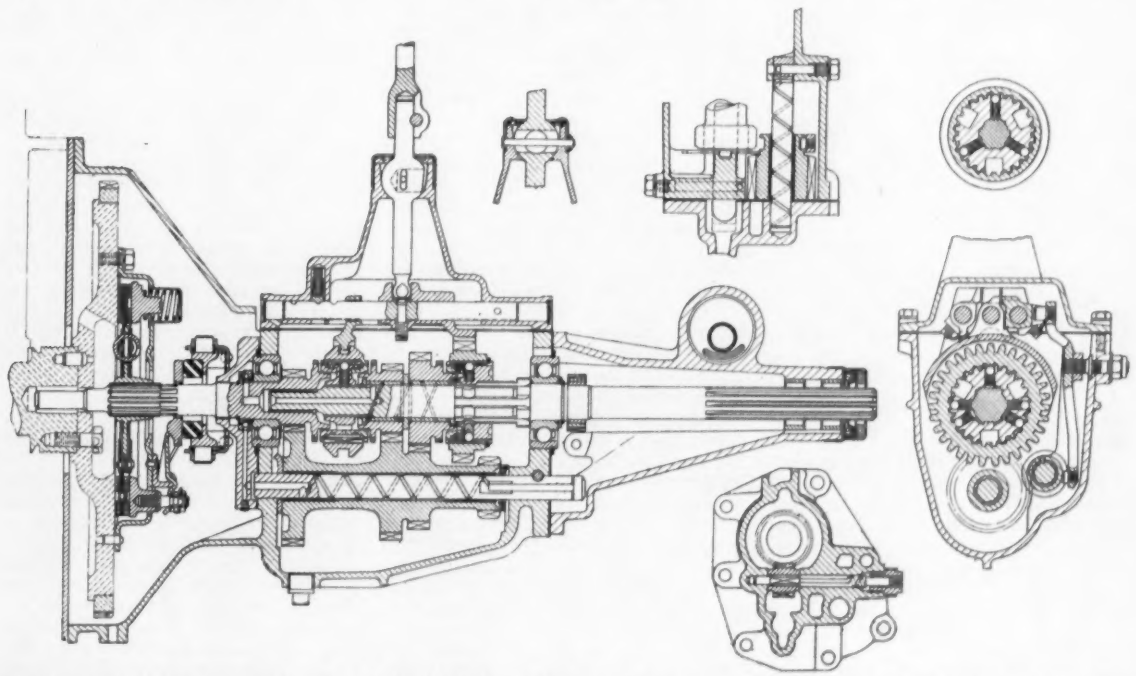
An En 352 mainshaft is employed. On the $\frac{7}{8}$ in diameter splined portion, immediately behind the front bearing, is the En 18C centre member of the synchronizer. Spring-loaded balls, housed in three radially-disposed holes spaced 120 deg apart in the centre member, register in a groove round the inner periphery of the En 18C sleeve member. The synchro cups are of



The clutch actuation mechanism

B.S.1400 PB1C phosphor bronze; they are carried in an annular space in the front and rear faces of the inner member. Three slots are machined longitudinally in the periphery of the inner member and three tongues on each cup register in them and allow only limited rotation of the cup relative to the inner member. The included angle of the cups is 12 ± 0·03 deg. Spiral grooves are cut in their faces to break down the oil film and to ensure that disengagement, for which positive mechanical provision is not made, is effected easily.

When a gear is selected, the sleeve and inner members are moved axially together to engage the cones. As a result, the cups rotate relative to the inner member, within the limits



In the gearbox a spiral groove machined on the layshaft feeds oil from a well in the rear extension casting to ducts in the front cover, and thence through holes in the mainshaft to the bushes for the gears

imposed by the freedom of movement allowed between the tongues on the cups and the slots in the member. Further motion of the sleeve overrides the detents, but it cannot pass through the teeth on the cup to engage the gear until synchronization has been effected, that is, until the tongues are centrally disposed in their slots. The leading edges of these teeth are chamfered to form a lead-in for the sleeve member which, when synchronization is complete, passes straight through and engages the coupling teeth on the gear selected.

On the $\frac{7}{8}$ in diameter portion of the mainshaft behind the synchro unit are the En 18C third and second speed gears, on each of which are machined the coupling teeth and inner synchro cone. These gears are mounted on a floating, chilled-cast, B.S. 1400 PB1C phosphor bronze bush. A 0·15 in thick flange round the centre of the bush separates the two gears. The inside and outside diameters of the bush are respectively 0·875 in and 1·092 in, and its length is 2·28 in. En 2 thrust washers, 0·125 in thick, are housed in recesses machined in the front and rear ends of the gear assembly. At the front, the thrust washer is retained by a circlip round the splines that carry the top and third synchro unit. The rear thrust washer is located against a shoulder where the shaft diameter is increased to 1·10 in and splined to carry the sliding first gear and second speed synchro unit.

Considerable attention has been devoted to the provision of adequate lubrication for the bushes. Oil is passed from the counterbore in the primary shaft, along an axial duct in the mainshaft and thence out through a radial hole to the front end of the bush for the third and second speed gears. From there, a spiral groove feeds it along the bush, and radial holes carry it out to other spiral grooves round the outer periphery of the bush.

An En 18C centre member for the second speed synchro unit is splined on the mainshaft in the rear of the box. Formed on the En 3A sleeve component is the first speed gear. The arrangement of the second speed synchronizing cone assembly, with its three spring-loaded balls, is the same as that for third and fourth speeds, but it also incorporates two extra balls housed together with two plungers in diametrically opposed holes in the inner member. When the sleeve member is slid into engagement with the second speed coupling teeth, it unseats the two balls from the detent groove in its inner periphery; these balls force the plungers inwards to register in an annular groove on the mainshaft, thus preventing the inner member from sliding to the rear when the gear is moved out of engagement.

When the selector forks are moved to the rear to engage bottom gear, the sleeve member comes against a snap ring round the centre member which it carries with it until the gears are engaged. Further rearward motion of the inner member is prevented by a collar machined on the shaft. The inner race of the rear ball bearing is immediately behind the collar. This race is assembled over the tail end of the shaft and held in position by a washer and circlip. The outer race is housed in the end wall of the gearbox and a snap ring in a groove round its periphery is clamped against the casing by the bolted-on rear extension.

GEARBOX DATA

	Pitch circle diameter	Helix angle	Diametral pitch	Material
Primary gear	1·500 in	30 deg L.H.	10	En 18C
3rd speed gear	1·900 in	30 deg L.H.	10	En 18C
2nd speed gear	2·500 in	25 deg L.H.	10	
1st speed gear	3·100 in	—	10	En 355
Layshaft cluster:				
Constant mesh gear	3·100 in	30 deg R.H.	10	
3rd speed gear	2·700 in	30 deg R.H.	10	
2nd speed gear	2·100 in	25 deg R.H.	10	En 355
1st speed gear	1·500 in	—	10	
Reverse idler gear	1·700 in	—	10	

A nylon speedometer gear is pressed on to the shaft behind the bearing. The spindle carrying its pinion has a longitudinal oil spreader groove on it. This groove communicates with an oil return scroll machined on the outer end of the spindle to prevent oil from being pumped up the cable casing to the instrument. About $3\frac{1}{4}$ in to the rear of this are the splines for the sliding joint. The sleeve member of the sliding joint is carried in two $\frac{5}{16}$ in long by 1·1870 in bore Clevite bushes housed in the tail end of the rear extension and spaced with their centres approximately $1\frac{1}{8}$ in apart. An oil return groove is machined longitudinally in the base of the housing for the bushes. A cupped steel housing pressed on to the tail end of the extension carries a lip type Super oil seal and a felt sealing washer. These bear on the sleeve member, which is integral with the companion flange for the universal joint.

A 0·655 in diameter En 32B layshaft is fitted. It is carried in the front and rear walls of the gearbox and positively located by the dowel end of a set-screw inserted from the side of the casing. The dowel end registers in a groove machined chord-wise in the periphery of the shaft.

The layshaft cluster is carried on two Clevite 10 bushes, one at each end. These bushes are 1·385-1·365 in long by 0·7815 in outside diameter by 0·6562 in inside diameter. Their bores are reamed to size after they have been pressed into the cluster. At the front end, a 0·125-0·123 in thick Clevite 10 washer is interposed between the cluster and the gearbox wall. Another Clevite 10 thrust washer is interposed between an En 2 washer, at the tail end of the cluster, and the end wall of the gearbox. The washers at the extreme ends are locked against rotation by pressing small portions of their peripheries into slots in the adjacent walls of the gearbox, while the steel washer at the rear is similarly locked to the layshaft cluster.

A noteworthy feature of the box is the lubrication system, which is arranged as follows. The rear end of the layshaft extends into a small well formed in the rear extension casting. From there, oil passes along a groove machined longitudinally on the shaft, into the rear end of the adjacent bush. It then passes along a spiral groove to the centre of the front bush. The groove terminates at this end at a radial hole that carries the lubricant into an axial passage in the shaft and thence out at the front, where it passes into the vertical duct in the front cover and up to the mainshaft lubrication system already described. The transfer of the oil along the length of the layshaft is effected by the rotation of the cluster round the spiral groove on the stationary shaft. The ends of the bore of the cluster are shouldered to take the bearings, so that the clearance between the bore and the shaft is small enough to effect the transfer of the lubricant.

A more usual method of lubricating the mainshaft bushes is to drill radial holes in the gears. This, however, is not entirely satisfactory since the tendency is for the oil to be flung outwards by centrifugal force rather than to pass inwards to lubricate the bushes. It would appear likely that, by taking extra care to provide effective lubrication, wear at the ends of the bushes and consequent tilting of the gears will be appreciably reduced. If this is so, the box should remain relatively quiet in operation for a much longer period than usual.

One end of the 0·56 in diameter, case-hardened En 32B reverse idler spindle is carried in a blind hole in the rear extension casting. The other end is in a boss on the side of the gearbox. Adjacent to the rear face of this boss is an annular groove machined on the spindle to communicate with a spiral groove that passes oil into the Clevite bush in the idler gear when in the engaged position. This bush is 1·79 in long and is pressed into the gear. The length of the teeth on the idler is about $1\frac{3}{16}$ in so that when it is slid forwards into engagement it meshes with the

mainshaft gear and the rearmost pinion on the layshaft cluster.

All three case-hardened En 32B selector rods are 0·4990-0·4982 in diameter and are carried in the top cover of the gearbox. Spring-loaded plungers, also of En 32B, are housed at the front in blind holes in the cover and register in grooves in the tops of the rods to form the usual gear lock arrangement. The gear interlock mechanism is at the rear ends of the rods. It takes the form of a ball between each pair of rods and a plunger housed in a horizontal, radial drilling in the centre rod. Movement of any one rod from the neutral position forces the balls into grooves in the other pair of rods and so prevents selection of more than one gear at a time.

The En 32B striker lever is in a turret housing cast integrally on top of the cover. Each of the B.S.1400 AB1C aluminium bronze selector forks is secured by a dowel ended set screw screwed radially into its boss and registering in a hole in the rod. Reverse gear is selected by a centrally pivoted lever. The pivot bearing is of the screw type, and the threaded En 1A pivot pin is carried in the side wall of the gearbox. A pin is pressed into each end of the lever to engage, at the top, in the selector fork and, at the bottom, in the groove round the front end of the reverse idler gear.

Reverse baulk is formed by a shelf on the centre selector fork, over which the lever has to be lifted. The striker lever is retained in its ball housing by a pin passed horizontally through an elongated hole in the ball. A rubber sealing ring is fitted over the top of the ball-socket and is retained by a pierced cupped pressing held in position over the top of the turret by means of the pin that retains the lever. The gear shift control lever has a socket in its base and is assembled on to a short extension of the striker lever. It is retained by a bolt, the shank of which registers in a groove machined in the periphery of the extension.

Back axle

A Hardy Spicer, needle roller jointed propeller shaft transmits the drive to the back axle. Its diameter and effective length are respectively 2 in and 44⅛ in. The axle is of the semi-floating, hypoid type with a banjo casing, and is mounted in a 1 deg nose-up attitude. A final drive ratio of 4·875 has been adopted. The oil capacity of the unit is 1½ pints, and S.A.E. 90 grade EP lubricant is specified. A B.S.310 malleable cast iron gear carrier is spigoted into the banjo casing and secured by means of eight $\frac{7}{16}$ in diameter set-screws locked by spring washers.

There are eight teeth on the crown wheel pinion. Its axis is offset ¾ in below the axis of the crown wheel, and $\frac{7}{16}$ in to the right of the plane containing the axis of the differential pinion spindle. The pinion is up-ended on its En 35A shaft, which is 1 in. diameter at the rear bearing and ¾ in diameter at the front. A turned, tubular distance piece between the inner races of the bearings spaces them about $1\frac{7}{16}$ in apart. One bearing is assembled from the front and the other from the rear into the nosepiece. Their outer races are located against shoulders in the housings. Shims in front of the outer race of the rear bearing control the preload. The whole assembly is pulled against the pinion by a nut on the ⅝ in diameter threaded end of the shaft. This nut is tightened against a plain washer and the companion flange for the universal joint, which is splined on. A Super oil seal, housed in the front end of the nose piece, bears on the boss of the universal joint flange and is shrouded by a cupped pressing mounted on the boss.

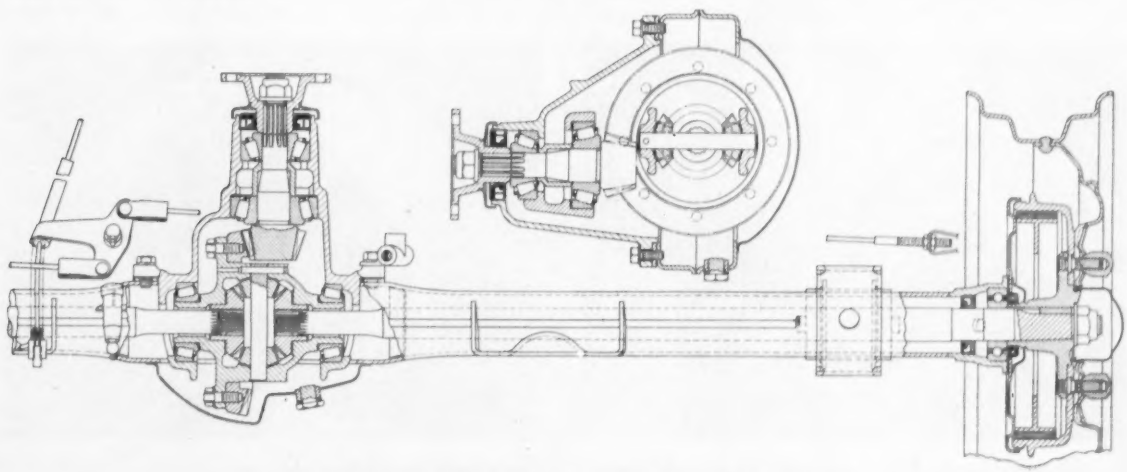
There are 39 teeth on the En 35A crown wheel and the backlash between it and the pinion is 0·004-0·006 in. The crown wheel is spigoted on to the differential cage and is pulled against the inner face of a flange by eight $\frac{7}{16}$ in diameter set bolts screwed into tapped holes in the wheel. Its overall diameter is 5·763-5·758 in. The one-piece differential cage is made of Lepaz malleable cast iron. A $\frac{7}{16}$ in diameter peg, pressed in a hole in the cage, and secured by peening, is passed through the end of the ½ in diameter, En 355 differential pinion spindle. Two En 355 differential pinions are employed. The length of tooth engagement with the differential wheels is approximately $\frac{7}{16}$ in, and the backlash is 0·004-0·006 in. A 1·13 in diameter phosphor bronze thrust washer is interposed between the cage and each of the spherical outer faces of the pinions.

Flat phosphor bronze thrust washers are interposed between the En 355 differential wheels and the cage. The overall diameter of these wheels is 2·286-2·289 in. Their bosses are 1⅛ in diameter and they bear for a length of $\frac{7}{16}$ in in the cage. The diametral clearance is 0·0022 in-0·0040 in.

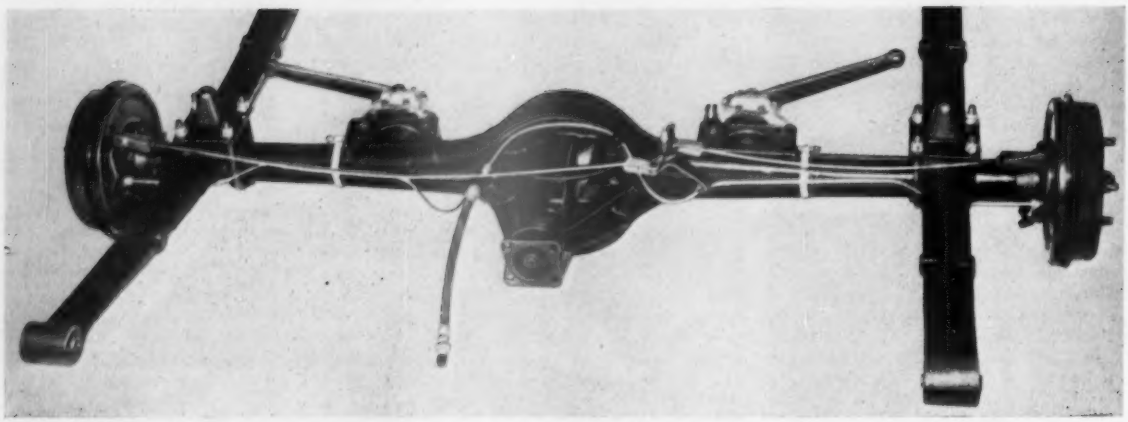
Two taper roller bearings, spaced with their centres approximately 4½ in apart, carry the cage. Their outer races are housed in the nose piece casting and are retained by bolted-on caps. A stiffening rib is formed round the outer end of each cap and abuts against a similar rib outboard of the housing. Shims are interposed between the inner races of the bearings and shoulders on the cage for adjusting both the preload and the meshing of the crown wheel and pinion.

At the inner end of each half shaft, where it is carried in the differential wheel, are 18 involute serrations. Their root diameter is 0·7043-0·7083 in and the depth is 0·0874-0·0794 in. A pressure angle of 45 deg has been adopted and the diametral pitches are 24/48. Immediately outboard of the splines, the shaft, where it passes through the differential cage, is machined to the spline root diameter. It is then increased to $1\frac{1}{8}$ in diameter. The shaft section is tapered to 1 in diameter near its outer end where it carries the inner race of the wheel bearing.

This is a single row, ball bearing, with the outer race carried in a flanged tubular housing butt-welded to the 2 in diameter by 10 S.W.G., En 2C axle



A banjo type, semi-floating rear axle with a hypoid pinion is employed



Rear axle and spring assembly, with the lever type shock absorbers mounted on it

tube. Inboard of the bearing, a Super oil seal bears on the shaft, and another is carried in a dished, pressed-steel housing clamped between the flange and the brake back plate. This seal bears on the end of the boss of the En 4 driving flange. The shaft is tapered one in twelve to receive the keyed-on driving flange, which is retained by a self-locking nut on the $\frac{3}{8}$ in diameter threaded end of the shaft. A B.S. 1452 grade 17 cast iron brake drum is spigoted on to the driving flange and secured by countersunk set screws. Four $\frac{3}{8}$ in diameter studs, together with conical seating, cap nuts, secure the wheel.

Rear suspension

Semi-elliptic leaf springs are employed in conjunction with Armstrong lever arm type DAS8R shock absorbers for the rear suspension. The springs are mounted on brackets welded on top of the axle tube and are peg-located, and the shock absorbers are on brackets welded to the rear face of the axle tube. Drop links connect the ends of the lever arms to 16 S.W.G. brackets welded to the longitudinal members that support the rear springs. Difficulties arising from the aeration of the shock absorber fluid due to the axle motion have not been experienced. Rubber bump stops are fitted on the spring retainer plates and in the full bump position bear on the longitudinal members, which are stiffened locally by 16 S.W.G. top hat section doubling plates.

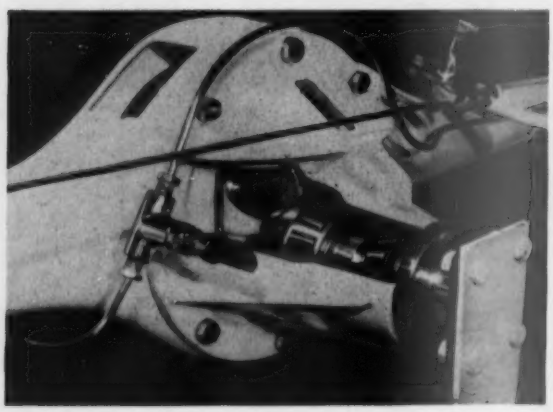
To help to give an understeering tendency, the axle tube is offset $3\frac{1}{2}$ in forward of the spring centre and the front spring eye is $4\frac{7}{8}$ in below the rear one. The unsprung weight is 128 lb. A periodicity of 86 cycles/min is obtained with a spring rate of 100 lb/in and a deflection of 4·75 in to the laden position. To the full bump position the deflection is $7\frac{1}{4}$ in.

The length of the spring with the vehicle in the static, laden position is $44\frac{3}{4}$ in between eye centres. Rubber bushes are fitted in the spring eyes and shackle pivots, and $\frac{7}{16}$ in diameter, En 18T pins are employed. The whole assembly on each side is carried in En 2C brackets on a longitudinal box-section member on the body structure. Four, 2 in wide leaves of En 45A are incorporated in each spring. The top one is 0·234 in thick, the next two 0·219 in and the fourth is 0·188 in thick.

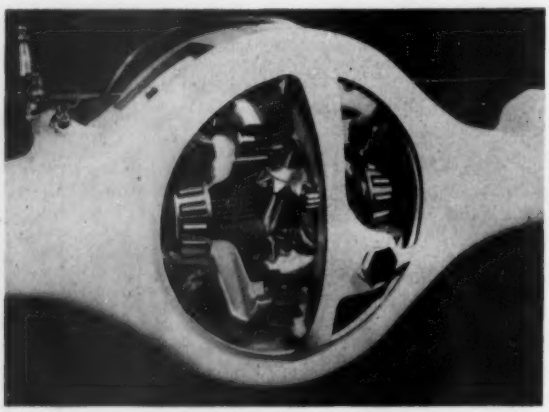
Front suspension

An interesting feature of the front suspension is the way in which it is mounted on the body structure. One of the most difficult problems to be solved when unitary construction is employed is the prevention of transmission of road noise to the structure. Various methods of solving this problem have been adopted by different manufacturers. The method adopted on the Standard Eight appears to be particularly well thought out. The double-transverse-wishbone-link type suspension units, together with the coil springs, Girling DAS4 telescopic shock absorbers and steering box are mounted on a cross member. On top of this member, which is of top hat section with a closing plate spot-welded to its flanges, two brackets are welded, one at each side under the side members of the front end structure. Carried in the front end of each bracket is a rubber bush with its axis positioned transversely. This bush is $\frac{1}{16}$ in outside diameter by $1\frac{1}{8}$ in long, and a $\frac{7}{16}$ in diameter bolt is passed through it to secure the unit between two lugs of a pressed steel bracket welded under the side member. At the rear end of each bracket is seated a conical rubber mounting incorporating a rebound rubber.

This mounting is in two pieces and its axis is vertical. The lower portion is the rebound rubber and is fitted



External gussets on the gear carrier stiffen the areas adjacent to the bearings that support the differential cage



Two differential pinions are employed, and the crown wheel is secured to the inner face of the flange round the cage

beneath a $\frac{1}{4}$ in thick washer welded under the bracket. It is spigoted into the hole punched through the washer and the bracket. The other portion of the mounting is carried above the bracket, and a dished pressing welded under the frame side member seats on top of it. The whole assembly is secured by a $\frac{7}{16}$ in diameter bolt passed from the top through the centre of the unit. This bolt is shouldered and a $1\frac{3}{8}$ in diameter washer is passed over its lower end and tightened against the shoulder by a self-locking nut.

When assembled, the overall depth of the two rubber components is $1\frac{3}{8}$ in and the diameter of the area on which each bears on the bracket is $1\frac{1}{2}$ in. The vertical axes of the two rear mountings are 1 in in front of the centres of contact of the wheels with the road, so they take practically all the vertical loading. Positive location in the horizontal plane is effected by the front pair of mountings. This feature is of particular interest because location of the suspension units is, of course, of vital importance so far as the steering characteristics of the vehicle are concerned. Rubber bushes have been employed in the front mountings because, in addition to acting as pivots to allow for deflection of the rear mountings, they also take drag loads and are therefore subject to vibrational impulses transmitted from the road.

FRONT SUSPENSION DATA

Castor angle	$+1\frac{1}{2}$ deg, laden
Camber angle, laden	+2 deg, laden
Camber angle change from full bump to rebound	2 deg 15 min
Track change from fully laden to full bump	$\frac{1}{2}$ in
Toe in	0-$\frac{1}{16}$ in
Swivel pin angle	7 deg
Height of roll centre above the ground	3 in
Angle of trail of the transverse links	25 deg
Unsprung weight	102 lb
Front end rate at spring	230 lb in
Front end rate at wheel	70·8 lb in
Deflection to laden position	3·8 in
Deflection to full bump	6·93 in
Spring details : Material	En 45A
Free length	11·44 in
Length fitted, fully laden	7·63 in$\pm\frac{1}{16}$ in
Coil mean diameter	3·75 in$\pm$0·02 in
Coil wire gauge	0·50 in$\pm$0·002 in
Number of coils	$7\frac{1}{4}$

The leading particulars of the geometry of the suspension are given in the table. Both the upper and lower wishbone links are fabricated from En 8Q pressings. The upper link is $7\frac{1}{4}$ in long between the bearing axes and the lower one is $12\frac{1}{2}$ in long. At the frame, the vertical spacing of the bearing axes is about $8\frac{1}{4}$ in. On the upper wishbone, the pivot bearing is 12 in from the centre line of the chassis and on the lower one it is $10\frac{1}{16}$ in from the centre line.

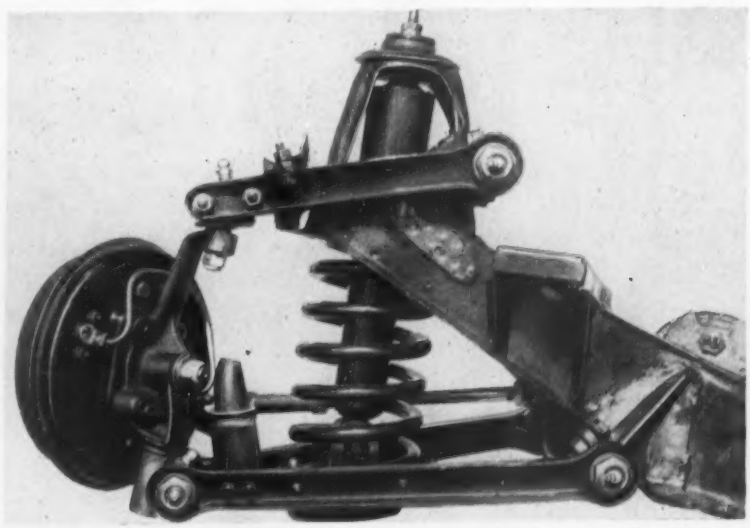
At each wheel, a ball joint carries the end of the upper link, and two screw-type bushes, with their centres $2\frac{1}{4}$ in apart, are mounted in the ends of the arms of the lower link. The vertical spacing between the centre of the ball joint and the axes of the screwed bushes is 9 in. From the centre line of the chassis the distance to the ball joint is 21 in, and to the axes of the bushes it is $21\frac{1}{16}$ in.

The pivot bearings of the upper wishbone arm are also of the screw type. On the pin, the thread at each end is approximately $\frac{3}{4}$ in long by $\frac{3}{8}$ in diameter. Flanged, En 2A bushes are screwed into bosses formed in the ends of the wishbone arms. Mild steel washers are interposed between the flanges and the outer ends of the bosses. The flanges are of hexagon form to take the spanner with which they are tightened against the washers. Sealing discs, with grease nipples screwed into their centres, are pressed into counterbores in the outer ends of the bushes and secured by peening over the peripheries of the counterbores. Each arm is screwed separately on to the end of the En 16R pin forging. This forging is bolted to a bracket inboard of the turret that houses the upper end of the shock absorber, which is mounted co-axially with the spring. Rubber sealing rings are interposed between the inner ends of the bushes and collars formed round the pin.

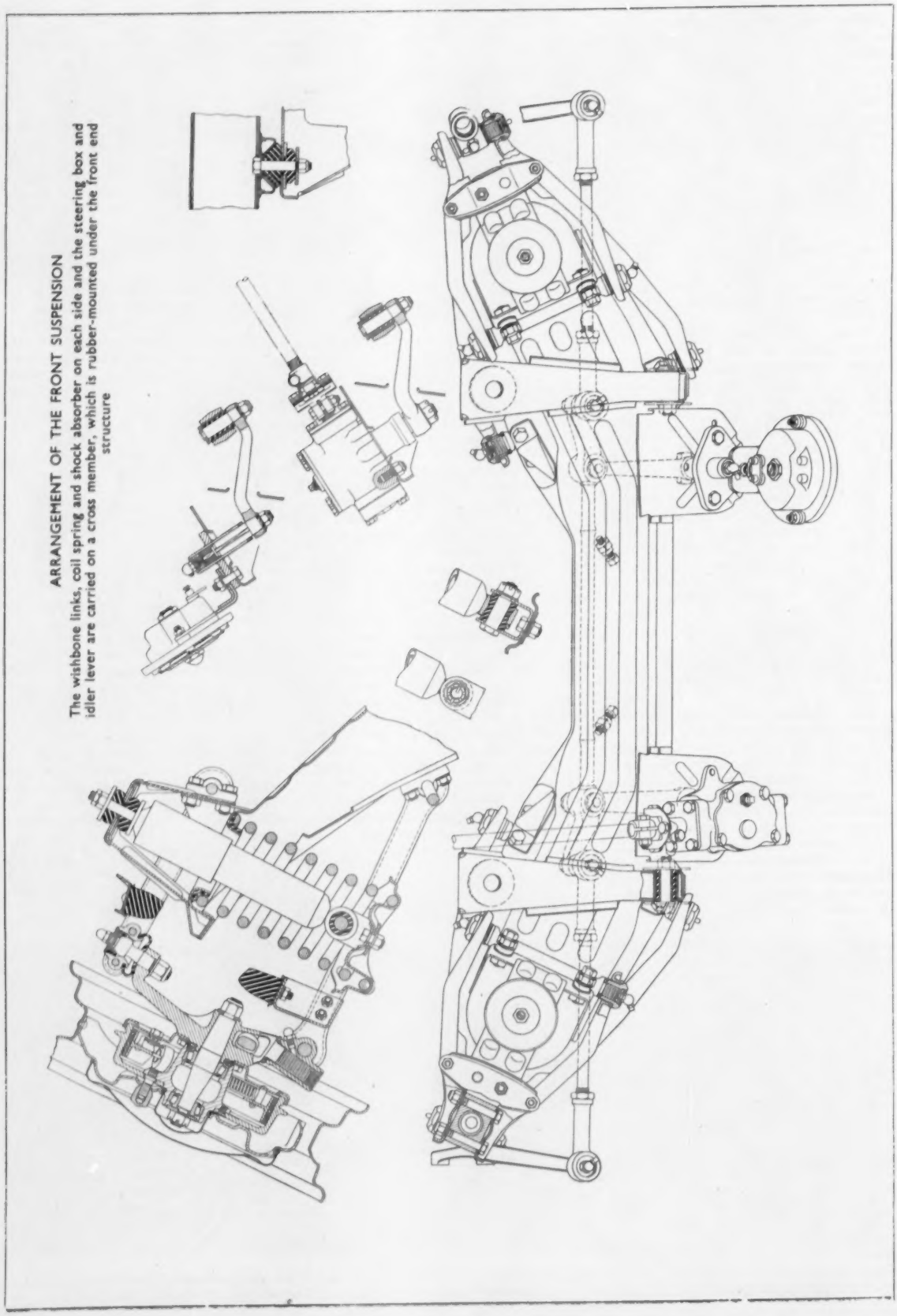
Between the outer ends of the arm, the ball and socket joint is secured by means of two $\frac{5}{16}$ in diameter bolts and self-locking nuts. Immediately inboard of this joint, a pressed steel bridge-piece carrying the rubber rebound stop is bolted to the arms. The diameter of the case hardened En 32B ball is $\frac{7}{8}$ in. Its socket is of En 3A and it is retained in position by an En 2A cupped-pressing, which is spring-loaded to provide automatic adjustment for wear. One end of the spring bears against a flange round the pressing and the other against the closing disc which is peened into a counterbore in the top end of the socket. A grease nipple is screwed into the centre of this disc.

Pivot pins for the lower wishbone link are formed on the ends of an En 16R stamping. This stamping is cranked downwards at the centre to clear the steering rod. It is secured by four $\frac{1}{4}$ in diameter bolts to the flanges at the base of the cross member; pressed steel gusset brackets in the angle between the flanges and the front and rear vertical faces of the cross member provide reinforcement at the attachment points. A screwed bush assembly, similar to that employed on the upper link, is fitted in the end of each arm of the lower link, but in this assembly the bushes are $1\frac{1}{4}$ in long by $\frac{3}{4}$ in diameter. Two more bushes are mounted in the outer ends of the arms of this link. They are $\frac{3}{4}$ in long and carry a $\frac{3}{8}$ in diameter pin on which is mounted the lower swivel pin bearing boss. Rubber sealing rings are interposed between the inner ends of the bearing bushes and the boss. Immediately inboard of the bearing, a bridge-piece carrying the bump stop is bolted to the wishbone arms.

The lower spring pan is bolted between the arms of the wishbone. A U-bracket, between the arms of which is carried the rubber bushed ring type fitting on the lower end of the shock absorber, is secured by a single $\frac{1}{4}$ in diameter bolt through its base to the centre of the spring pan. The pin that carries the shock absorber is $\frac{3}{4}$ in diameter. At the upper end of the spring, the seating is pressed in the upper component of the cross member

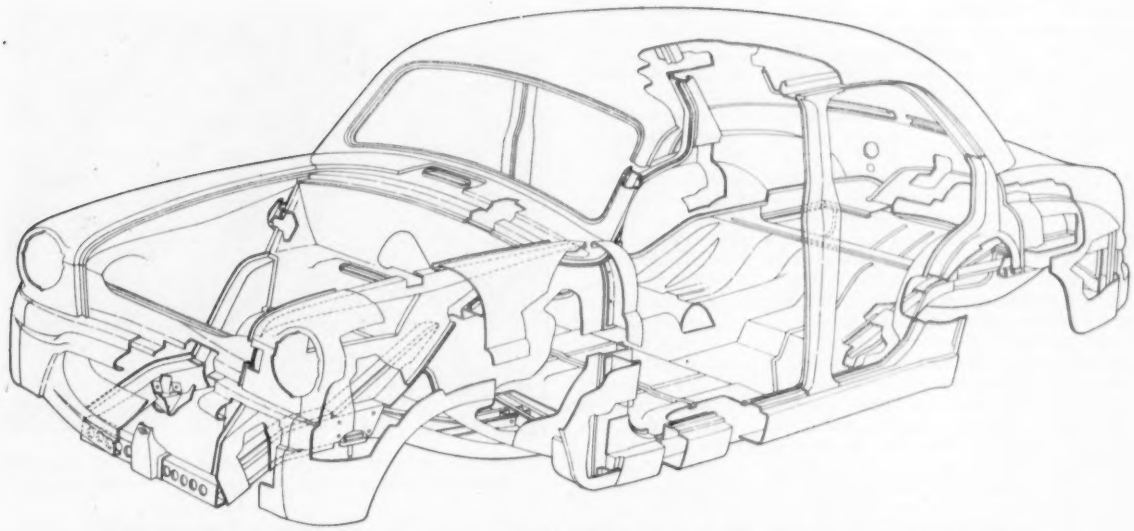


An unusual feature of the front suspension is the arrangement of the steering arm which is passed through a hole in the forged vertical link that carries the stub axle



ARRANGEMENT OF THE FRONT SUSPENSION

The wishbone links, coil spring and shock absorber on each side and the steering box and idler lever are carried on a cross member, which is rubber-mounted under the front end structure



Many components of the body-chassis structure are bolted on and can be readily replaced in the event of damage

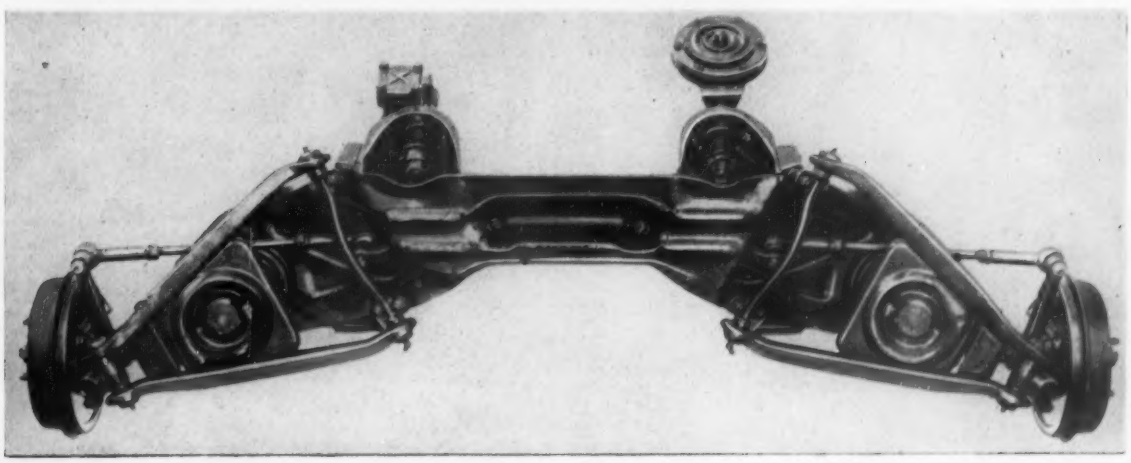
supporting the suspension. Interposed between this end of the spring and its seating is a $\frac{1}{8}$ in thick rubber ring. It is of interest to note that no rubber ring is fitted between the spring and the lower pan which is shaped so that there is no dead coil at the base of the spring.

A pressed steel turret is welded on top of the end of the cross member and a $2\frac{1}{4}$ in diameter hole is punched in its crown. Two large diameter, dished steel plates are fitted, one above and the other below the crown of the turret. They have $\frac{3}{4}$ in diameter holes punched in their centres and spigoted into them are the two rubbers of the sandwich type fitting at the upper end of the shock absorber.

To carry the lower end of the swivel pin, a $\frac{3}{4}$ in diameter, screw type bearing is machined directly in the boss between the arms of the lower wishbone link. The bottom of this bearing is sealed by a disc peened in a counterbore. A $\frac{3}{16}$ in diameter hole is drilled in the centre of the sealing disc to relieve any tendency for a hydraulic lock to develop beneath the swivel pin. The effective length of the bearing on the pin is $1\frac{5}{8}$ in. A rubber sealing ring is interposed between the upper end of the boss and a shoulder on the swivel pin forging. Lubrication is effected through a grease nipple screwed into the boss.

A forged En 8Q, vertical link pulled on a taper on the stub axle serves as a swivel pin fitting. An unusual feature is that a hole is formed in the forging at a point just below the axis of the stub axle, and the steering arm is passed through it. The arm is secured by two $\frac{3}{8}$ in diameter bolts spaced with their axes $1\frac{1}{16}$ in apart, one in front and one behind the vertical plane in which is the wheel axis. This arrangement undoubtedly contributes much to the overall rigidity of the steering system.

The En 8Q stub axle is pulled on to a 1 in 8 taper on the diameter of the swivel pin forging. It is secured by a self-locking nut on its $\frac{1}{2}$ in diameter threaded inner end. The brake back plate is spigoted and bolted to the outer face of the vertical link. Two taper roller bearings carry the B.S.310 malleable cast iron hub. They are spaced with their centres approximately $2\frac{1}{16}$ in apart. The inner race of the inner bearing is carried on a 1 in diameter portion of the stub axle, while that of the outer bearing is on a $\frac{7}{8}$ in diameter portion. These bearings are assembled into the hub from each end and their outer races bear against shoulders in their housings. The whole assembly is retained by a slotted nut and plain washer on the $\frac{1}{2}$ in diameter end of the stub axle. In the event of failure of the bearings, the wheel cannot come off the axle because the washer is of such a diameter that it will not pass through the outer race of the bearing. A hard felt, grease seal is carried in the inner end of the hub and bears on a shoulder round the vertical link. Round the hub is a flange



In this view, from below the front suspension sub-assembly, it can be seen that the track rod passes through the front cross member

that carries the four $\frac{3}{8}$ in diameter studs which, together with conical seating cap nuts, carry the wheels. The brake drums are spigoted on to the flange in the usual manner, and secured by countersunk set screws.

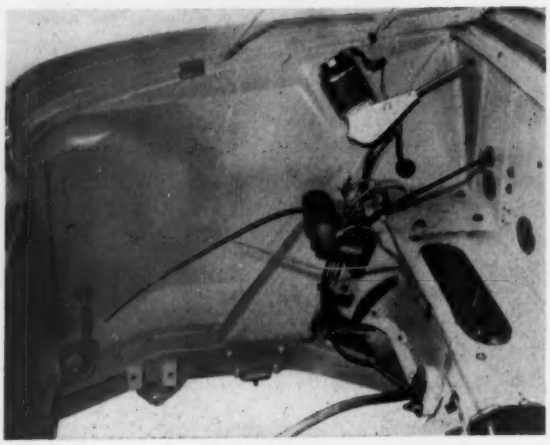
Steering

Burman worm and nut steering gear is employed. The steering box is mounted on a bracket on the cross member that carries the front suspension. Thus, road shocks transmitted to it are insulated from the body structure by rubber mountings between that member and the side members. The steering ratio is 14:1 in the straight ahead position and 12·7:1 on full lock. This gives $2\frac{1}{4}$ turns from lock to lock of the 16 in diameter steering wheel. The attitude of the inner steered wheel at full lock is 30 deg while that of the outer one is 28 deg from the straight ahead position. A turning circle of 32 ft is obtained.

A three-piece track rod layout has been adopted, with the centre rod housed inside the front cross member. The effective length of the En 18S drop arm is $4\frac{1}{2}$ in. Similar brackets are provided for both the steering box and the idler lever assembly, so that conversion for left-hand or right-hand steering is a simple matter. The idler lever is pulled on to the tapered end of its spindle by means of a self-locking nut. A screw thread is cut in the upper $\frac{1}{2}$ in of the end of the spindle, the overall length of which is $3\frac{3}{4}$ in. This end is screwed into a bearing bush formed by a $\frac{7}{8}$ in diameter steel tube threaded at one end and welded in a hole in a $\frac{1}{4}$ in thick plate. The plate forms a flange by means of which the assembly is bolted to the bracket on the cross member. Sealing at the upper end of the tube is effected by a steel disc peened in a counterbore and a grease nipple is screwed into the centre of the disc. A rubber sealing ring is housed in a groove in the bore at the lower end of the tube.

An eye-fitting is butt-welded at each end of the $\frac{7}{8}$ in diameter centre component of the three-piece track rod and in each is a rubber bush on a $\frac{1}{2}$ in diameter pin. These pins are mounted one on the end of the drop arm and the other on the idler lever. Their ends are tapered and threaded $\frac{1}{4}$ in diameter for self-locking nuts that pull them into tapered holes in the levers. A $\frac{1}{8}$ in thick washer is spigoted and peened on to the upper end of each pin to form a head to retain the bush and eye assembly. Adjustable stops are mounted on brackets under the cross member to limit the angular travel of the drop arm and idler lever.

The ends of the $\frac{1}{2}$ in diameter tie rods are threaded to take the outer components of the ball and socket joints by means of which they are attached to the drop arm, idler lever and the steering arms on the vertical link. When adjustment has been effected, the setting is fixed by means of lock nuts tightened against the end fittings. Grease nipples are provided for the lubrication of each of the ball joints which are self-adjusting for wear.

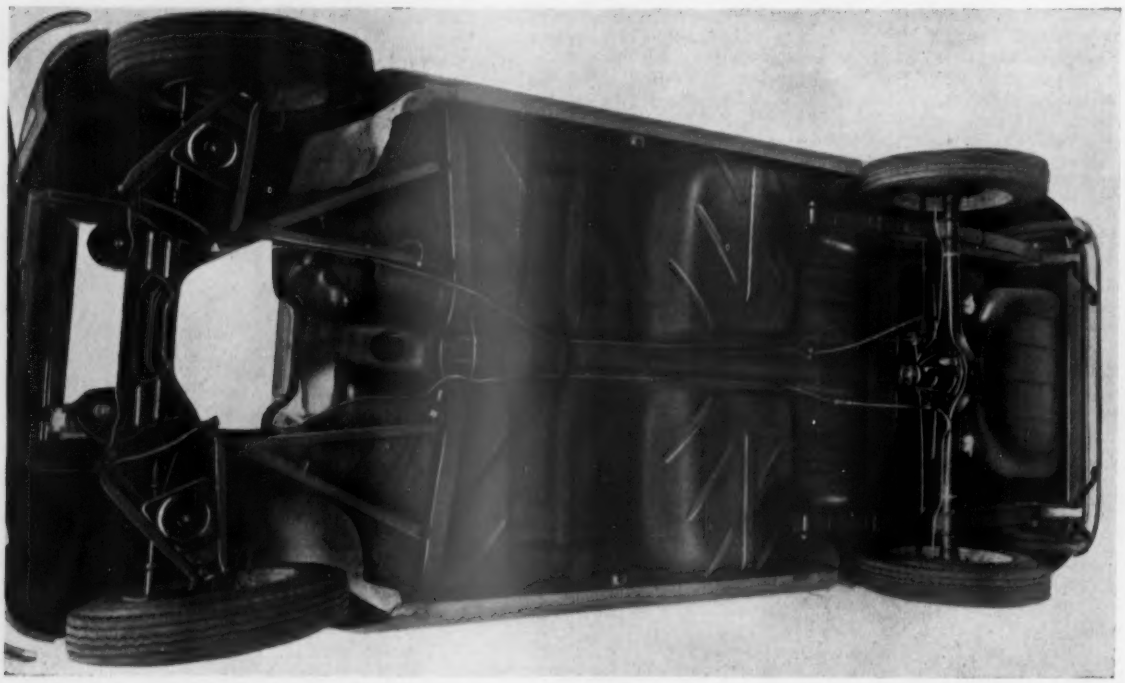


A view of the under-bonnet structure, showing the mounting of the brake and clutch master cylinder and reservoir units

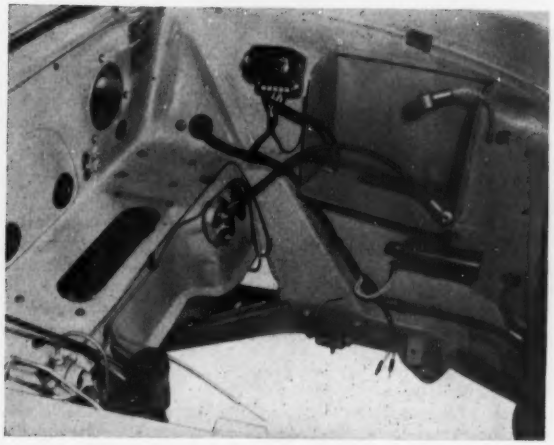
Brakes

Girling hydraulic brakes are employed. At the front they are of the two leading shoe type and at the rear one leading and one trailing shoe are fitted. Both the front and rear brakes are 7 in diameter and the shoe width is $1\frac{1}{4}$ in. The total friction lining area is 68 in^2.

Oilite bushes, 0·6265-0·6260 in inside diameter by 1·245-1·240 in long, carry



The rear engine mounting is carried between lugs welded to the cross member at the junction of the toe board with the front floor



The two mounting points for the suspension cross member can be seen from this view of the under-bonnet structure

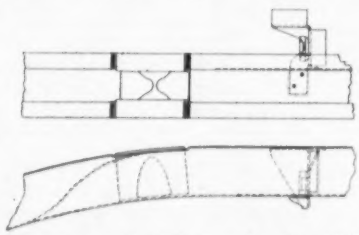
the pendant pedals of the clutch and brake controls. These pedals are pivoted on a common pin between two lugs welded to the front end structure. The brake pedal lever ratio is 4·22 : 1 and the total travel is 5·95 in.

A pull-up type hand lever, with a thumb-button ratchet control in its end, is centrally mounted on the floor. The lever ratio is approximately 5 : 1. Pinned to the lower end of the lever is a U-shackle to which is attached the threaded end fitting of the cable. Adjustment to the system is effected by means of two lock nuts screwed on the end fitting and tightened against the U-shackle. The cable is passed through a tube welded to the propeller shaft tunnel. This tube is sealed at its rear end by a felt washer held in position by a washer and a coil spring, the other end of which bears against a clip on the cable. At the rear end, the cable is attached to a pivoted and swinging link type compensator. Screw type bearing bushes, of En 3B and 1½ in long by 0·56 in inside diameter, are employed for all the pivots in the compensator. The pivot pin is of En 2. Cables are used instead of tie rods between the compensator and brakes. Fixed U-shackles on the inner ends of the cables are pinned to the lever; an adjustable U-shackle arrangement, similar to that at the hand brake lever, is fitted at the threaded outer ends of each, where they are attached to the brakes.

Body structure

Except for the side members of the front end structure and the dash brace, which are of 16 S.W.G., the 18 S.W.G. channel-section cross member at the extreme front end, and the 18 S.W.G. inverted top hat section cross member under the toe board, all the main components of the body structure are of 20 S.W.G. sheet steel. Between the heel board and the dash, the structure comprises two side members formed by the 2⅞ in wide by 6 in deep sills and the front pillars. It is reinforced by the centre pillars and rocker panels, together with two cross members. The front cross member of this part of the structure is of inverted top hat section, the flanges of which are spot welded to the floor. At the rear, the cross member is formed by the heel board and is divided at the centre by the propeller shaft tunnel. Additional stiffness is afforded to the floor by pressing it to form a transverse rib of channel-section under the front seats and over the propeller shaft tunnel. An inverted top hat section transverse stiffener is spot welded under the rear floor on each side of the tunnel.

At the front, two top hat section side members, 2 in wide by 4¼ in deep, are bolted through their flanges to the inner panel, or valance, of the wheelarch. A channel section cross member, 1⅛ in over the flanges by 2½ in deep, is bolted on between their forward ends. Right angle brackets of ⅜ in thick, mild steel plate are bolted on each side at the front to back up the bumper irons. Further back on the side members are the brackets for the suspension cross member. The rubber cone-mountings for this suspension assembly are immediately under the dash brace, which is a pressing so shaped that when spot welded to the wheelarch and dash, it forms a box section member to take most of the vertical loading.

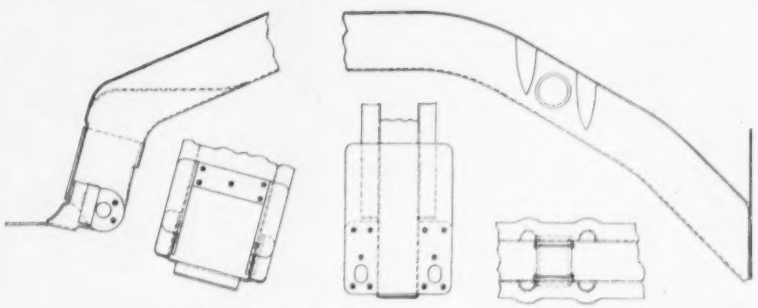


Bump stop and shock absorber drop link attachment near the centre of the rear longitudinal members

A bolted-on, Y-shaped component forms the continuation of each side member to the rear of the cone mounting. This part of the front structure is detachable so that access can be gained to the starter motor and other components for servicing purposes. The stem of the Y is inserted into the box section side member, and its arms are bolted to brackets welded to the cross member under the toe board. One of these brackets is beside the propeller shaft tunnel and the other is adjacent to the sill. They are formed by 16 S.W.G. U-sections inserted horizontally through holes punched in the walls of the top hat section cross member. The sides of the holes in the cross member are flanged and spot welded to the U-section, the base of which is at the front. Another 16 S.W.G. U-section is wrapped round the front end of the first one, but it is positioned vertically with its base at the bottom and its arms spot welded to the sides of the other, thus forming a box open only at the top. The holes for the attachment bolt of the Y-piece are punched through the two 16 S.W.G. thicknesses on each side of the bracket assembly.

Practically all the main components at the front end are readily detachable and therefore can be replaced by new ones in the event of their being seriously damaged. Included in these detachable components are the front wings and the grille panel. Another unusual and equally noteworthy feature of the vehicle is that the rear wing panels are also bolted on so that they can be readily replaced.

As has already been mentioned, the rear spring on each side is mounted on a longitudinal, top hat section member spot welded to the seat pan and boot floor between the heel board and rear skirt panel. The methods of reinforcing the spring eye and shackle attachments, as well as the bump stop attachment and shock absorber mounting brackets, are shown in an accompanying illustration. At the extreme rear end, the sides of the top hat section are flanged outwards and a 16 S.W.G. closing plate is spot welded on. On each side, a hole is punched through the flange and the closing plate. These holes are for the bolts securing the bumper iron, which is of cranked form.



The attachment of the ends of the rear longitudinal members, and the spring eye and shackle mountings

VACUUM METALLIZING

A Decorative and Protective Bright Metal Finish

THIS method of depositing a thin film of metal on a wide range of materials, including metals, plastics and glass, provides an economical alternative to electroplating for articles that are not subjected to hard or abrasive wear. It is applied over a coat of synthetic lacquer which covers up small surface irregularities and serves as a key for the metal film. By this means the need for preliminary polishing operations is eliminated. Though primarily regarded as a decorative finish it also has a protective value since the metal coating, in conjunction with the base lacquer, is continuous and effectively covers small-radiused or sharp edges. The thickness of the metal film ranges from 5 to 10 millionths of an inch only and usually it is protected against handling and wear by a coating of clear lacquer. Each lacquer coating, applied by spraying or dipping, is air-dried or cured. Such composite coatings are resistant to exposure to normal atmospheres and have undergone the usual 100 hr. salt-spray test without attack.

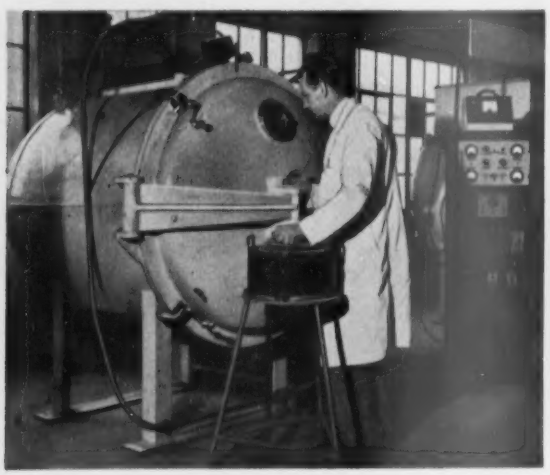
The process is based on the fact that metals can be vaporized in a vacuum. It is carried out in large sealed chambers that are exhausted down to 10^{-4}mm Hg by an eccentric vane pump and two diffuser pumps. (Normal atmospheric pressure is 760 mm Hg). Small articles to be metallized are mounted on the spindles of a support frame that can be slid from a light trolley directly into the vacuum chamber. All spindles are connected by chain drives to a central spindle that engages a hand crank in the rear wall of the chamber to enable them to be rotated to expose all sides of the articles to the sources of metal located centrally in the support frame. These consist of lengths of aluminium wire loosely wound around tungsten heating elements.

When the loaded chamber has been exhausted to the desired high vacuum the operator applies current to the heating elements and the aluminium wire is heated to incandescence and finally evaporated. The operator watches the process through a window in the chamber cover, avoiding the use of more current than is necessary to vaporize the aluminium, since vaporization of the tungsten will cause discoloration of the metal film. A second operator slowly rotates the spindles carrying the articles being treated. The vaporized metal spreads in all directions, forming on the articles a brilliant, lustrous film.

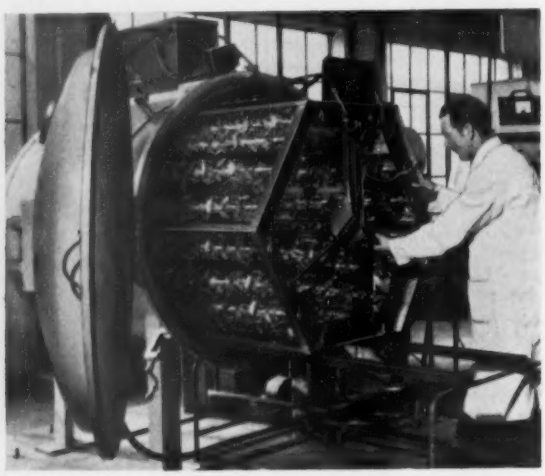
The process is most suitable for articles of small to medium size, say from 1 in to 8 in diameter. Very small and also large articles are more difficult to mount in the chamber to obtain satisfactory coverage and may necessitate special fixtures. The chambers used in the plant operated by A.E.R. (1938) Ltd., 196, Great Cambridge Road, Enfield, Middlesex, as shown in the illustrations, can accommodate articles to a maximum size of 3 ft by 3 ft or flexible sheet of 3 ft by 1·5 ft. Shape is not of prime importance as the coating is thrown into recesses and intricate figured surfaces. It is not practical, however, to coat the interiors of small bore tubes.

Normally, aluminium is the metal selected for deposition since it is economical, resembles a brilliant silver in appearance and is non-tarnishable. Other metals such as chromium, silver, gold or copper may be used for special applications. As regards colour the process is extremely flexible. By introducing a suitable dye in the protective lacquer coating a brilliant gold, red, green or other colour may be obtained. As a variation of the brilliant finish a metallized crackle effect can be produced on metals and phenolic plastics.

The process is admirably suitable for headlamp reflectors which, being totally enclosed, are not given a protective coating of lacquer in order to take full advantage of the brilliant metal surface. The reflectivity of bright aluminium is superior to chromium plate and is approximately equivalent to that of silver plate, with the additional advantage that it is non-tarnishable. An interesting application is a driving light, moulded in a plastics material, with the reflective coating deposited directly on the body moulding. The metallizing of "three-dimensional" mouldings in Perspex or other transparent plastics for car badges and horn buttons is well established. It could be utilized for various items of interior trim, and die-cast interior door handles, metallized and protected by a special lacquer, are currently under investigation. Red reflectors are metallized over a dyed base lacquer on the rear surface of transparent plastics mouldings and by suitable masking a clear disc or panel can be left so that a single component can serve for a reflector and a braking or direction indicator. Similar methods are used for signs, nameplates and such articles produced by vacuum forming. In that process a sheet of cellulose acetate is embossed by atmospheric pressure after being clamped over a mould from which the air is exhausted.



Vaporization of the coating metal is observed by the operator controlling the heating current



Withdrawing the loaded support frame after deposition of the metal and relief of the vacuum in the chamber

SHELL MOULDS AND CORES

Important Developments in Polygram Equipment

NOT surprisingly, in view of its potentialities, the past few years have seen greatly increased interest in the shell moulding process. As a method of economically producing relatively small castings of high quality and to close dimensional tolerances, the process is already well established. More recent developments have made it possible to use the process for castings up to more than 200 lb weight. Metals ranging from aluminium to stainless steel are cast successfully.

Polygram Casting Company Limited, Power Road, Gunnersbury, London, W.4, have been pioneers in developing both the actual process and the equipment to be used. Initially, the equipment for the Polygram process was designed for hand operation, but it was realized that mechanized equipment was essential if the process was to be exploited to the full. After considerable research, the Polygram mould maker Mark IV was introduced late in 1952. This machine embodied two distinctive features, multiple pattern plates, which by automatic indexing could be invested in rotation, and automatic springless ejection. Since then there have been important developments, which are dealt with in these notes.

Before the mechanical developments are discussed, it may be advisable to make brief reference to the materials for the Polygram process. These materials have all been specially developed for the process and have been fully tested in actual foundry practice. They include:—

Resin binders. The resin binder, when mixed with selected sands, provides the basic raw material for shell moulds. A large percentage of the Polygram resin binder consists of 100 per cent phenol resin, since this is considered to be more economical in the long run than the cheaper cresylic or urea resins. The remainder of the binder consists of patented ingredients introduced to give the optimum combination of stripping qualities, absence of peeling, mould strength and surface finish.

Conditioning agent. A new pattern plate must be treated with conditioning agent before it is put into use to ensure free separation of the moulded shell from the plate. This object is effected by a few applications of the Polygram conditioning agent, which also provides an effective base for the stripping agent.

Stripping agent. This is applied to the pattern plate before every shell mould to ensure ease of ejection. Both this agent and the conditioning agent contain no water, so there is no rusting of the ejection mechanism. On the contrary, Polygram materials tend to form a very fine skin of inert chemicals over the equipment that gives protection against corrosion.

Anti-separation agent. This operates to prevent the separation of a properly blended resin and sand mixture into its component parts.

Adhesives. The adhesives are used to seal the half moulds into one whole ready for the pouring. Polygram adhesives have been specially developed for the very quick setting that is required in assembling moulds on the Polygram shell mould closer, which is described later.

Mould makers

The latest Polygram developments for shell mould making are three two-station, pneumatically operated, semi-automatic machines with all electric heating. These machines, which are designated Mark V.E, Mark VI.E, and Mark VII.E, are fundamentally similar in design, the only major differences being in the size of pattern plates each machine will accommodate. The V.E machine will accommodate a range of pattern plates from 16 in × 12 in to 20 in × 14 in, the VI.E 14 in × 18 in to 22 in × 24 in and the VII.E 24 in × 20 in to 30 in × 36 in. A Mark VI.E machine is shown in Fig. 1. The following description applies to all three machines.

These machines have been specially designed to give long and efficient service under the extremely arduous conditions that are inseparable from foundry work. They are, therefore, very rigidly constructed. Equal importance has been given to the reduction of operational fatigue to a minimum. For this reason, the pattern plates are pneumatically put on to and taken off the dump box. Furthermore, the centralized bin unit minimizes operator movement.

Another refinement is that suitable counterbalances ensure that the heavy pattern plate comes gently into position on the dump box. Once the plate is in position on the dump box, it is automatically and positively clamped so that there is no escape of sand and resin mixture. These machines are kept to a minimum overall size through the arrangement by which mould ejection takes place within the area of the curing oven instead of on separate tables. Another important element of the design is the pin method of ejection, see Fig. 2, without there being recourse to springs to prevent the pins from dropping forward as the pattern plate is inverted.

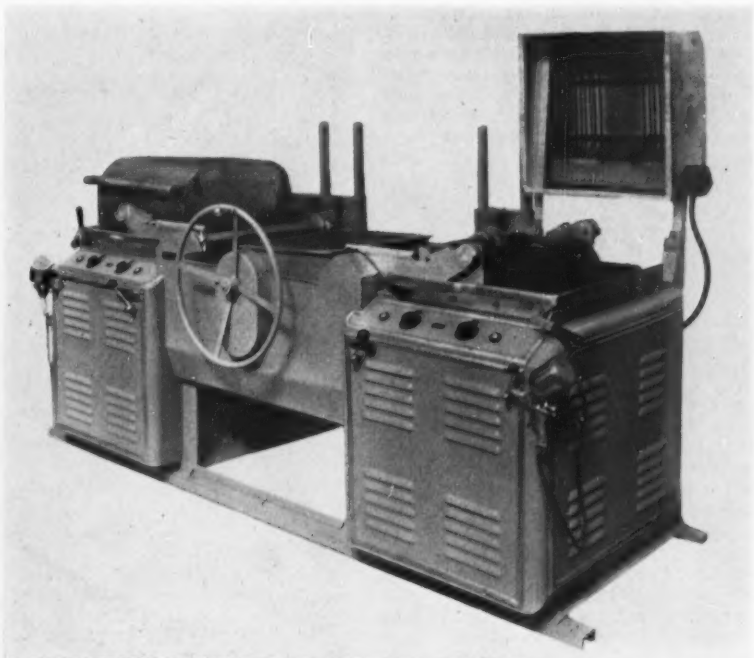


Fig. 1. Polygram semi-automatic shell mould maker Mark VI.E. 20 in × 24 in

The operation sequence is:—

(1) Heat the pattern plate by means of the electric elements situated with the oven above the pattern plate and in a static position below the plate.

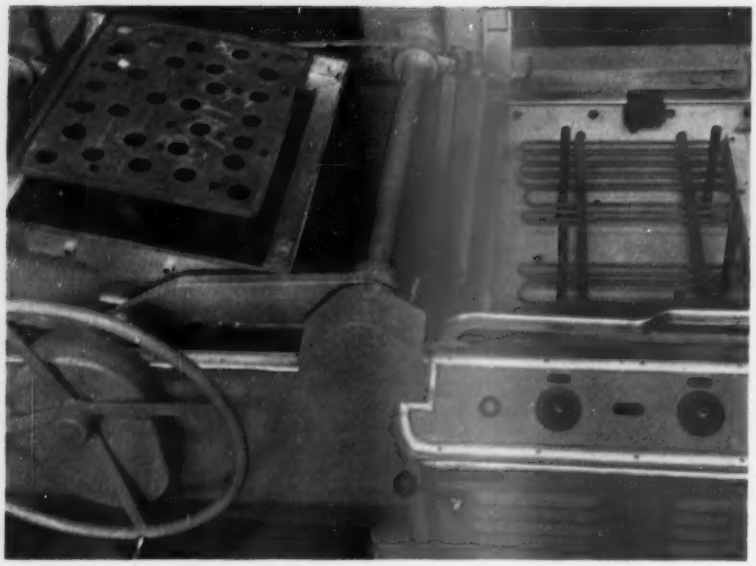


Fig. 2. The springless ejection system on the machine shown in Fig. 1

(2) Spray with stripping solution.
(3) Bring the pattern plate on to the dump box by means of the pneumatically roll-over device.
(4) Manually roll over the dump box to invest the pattern plate.
(5) Predetermined investment period.
(6) Return the dump box to its normal position.
(7) Operate the roll-over device to bring the pattern plate, with the soft plastic mould adhering to it, back to the original position.
(8) Lower the oven over the pattern plate and leave for the curing period.
(9) Lift up the oven and eject the mould from the pattern by means of the pneumatically - operated ejector plate.

During the curing operation at one station, the operator invests the pattern plate for the other station. By the time investment is completed and the second plate has been returned to the position for curing, the first mould will be ready for ejection.

Core makers

There can be little doubt that the Polygram core makers are in many respects the most important of the developments described in these notes. It was realized that a shell core would provide a high degree of accuracy, surface finish and permeability, and at the same time eliminate the need for core driers and stoving. Furthermore, a shell core maker would have a much wider appeal than shell moulding machines, since the use of shell cores need not be confined solely to shell moulding. On the contrary, they can also be used with advantage in conventional sand moulding and in gravity die-casting. From the outset it was considered desirable that the machines should be suitable for a wide range of both simple and complex cores, and that the cores produced should, wherever possible, be hollow to give high permeability and low material cost; at the same time, the cores must have high strength, good delineation and high surface finish. As a result of several years of research and testing, three Polygram core makers have been developed; the Mark VIII.E fully-automatic machine shown in Fig. 3, the Mark IX three-station bench-type machine and the Polygram-Croning Mark XI machine.

The Mark VIII.E machine has been designed specifically for large quantity production of similar cores. Essentially it comprises a filling and evacuating head, and is a two-station unit fitted with controlled, electrically - heated closing and stripping clamps. Each station will hold a pair of boxes, and the maximum size of box that can be accommodated is 9 in × 6 in × 6 in.

Sand/resin mixture is carried in a hopper of ample capacity that can be rotated through 360 deg, so that cores too large to be accommodated in the grips can be made by suitably rotating the head. The core boxes are opened and closed pneumatically by a simple lever control, and precise location is automatically assured. The electrically-heated, Sunvic controlled, clamping jaws maintain the core boxes at the correct investment temperature. When an appropriate push button is depressed, sand/resin mixture from the hopper fills the box at the required pre-set pressure. After investment is complete, another push button is operated and any excess material is evacuated to a storage bin for subsequent use. An efficient mechanical seal prevents loss of expensive material during the filling operation. A built-in pyrometer, spray gun and air blast make this machine completely self-contained.

The method of operation is:—

(1) The core boxes are brought to the correct operational temperature and sprayed with stripping solution.
(2) The hopper head is moved over the core box and clamped into position; a push button is depressed and the sand/resin mixture is rapidly forced into the box.
(3) After the five seconds investment period, another push button is depressed and the surplus material is evacuated.
(4) The hopper is positioned over the second box and the cycle is repeated.
(5) Shortly after the second box has been filled, the first core will be fully cured and the box can be returned to the starting position and the core lifted out.

There can be no doubt that this machine has tremendous potentialities for long run production. In the first place, it can be used for complex cores that hitherto have been produced by hand ramming. Secondly, the total production time per core is much lower than with any method previously employed. The actual filling time is

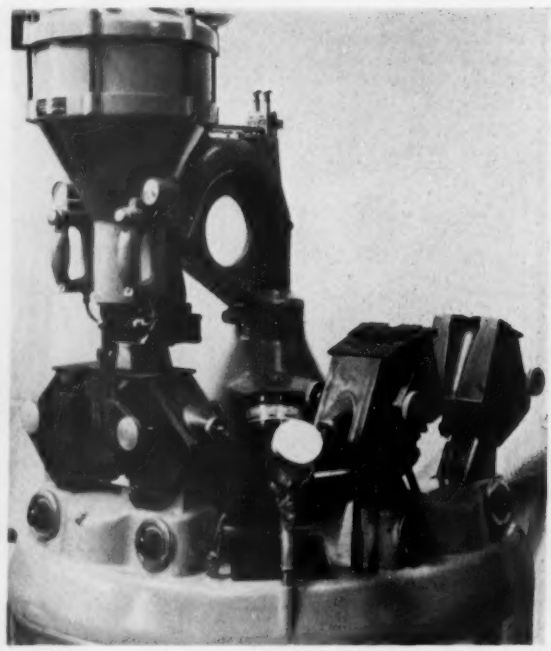


Fig.3. Polygram automatic shell core maker Mark VIII.E

appreciably less, but the greatest time saving arises from the very short investment period as compared with normal core baking. Finally, the automatic evacuation of surplus material allows the maximum effective use to be made of the sand/resin mixture and also maintains an extremely clean working atmosphere.

The Mark IX machine, see Fig. 4, has been designed to accommodate a much wider range of core boxes than the Mark VIII.E. It has the same design of blowing head for filling the boxes and evacuating the surplus material, but in place of the fixed jaws for clamping the core boxes, there are three tables that can be adjusted to suit the varying heights of three core boxes to be operated in turn. It also differs from the Mark VIII.E in that it must be used in conjunction with a separate oven for heating and curing purposes. Many conventional sand foundry core boxes can be used on this machine without alteration, providing they are made of a suitable metal. The operation cycle is:—

(1) A core box, previously heated to the working temperature in a separate oven and treated with the stripping solution, is held in position on the work table. The blowing head is then brought into position to fill the box and then evacuate the surplus material.
(2) The filled box is returned to the separate oven for curing.
(3) While the first core is being cured, a second core is made on the second table, which may be adjusted to take a different height of core box.
(4) A third core is made in the same manner at the third station, which will accommodate boxes up to a height of 24 in.

The Polygram-Croning Mark XI machine, see Fig. 5, has been developed specially for jobbing work. It is therefore designed to deal with a wide variation in type of size of core. This core maker differs from the two previously described in that it does not embody an automatic evacuation system. The operation cycle is:—

(1) A core box, previously heated in a separate oven and treated with the stripping solution, is positioned over the blowing head and held by an adjustable clamp.
(2) Operation of a pedal admits the sand/resin mixture. A pressure control valve allows any desired blowing pressure to be applied. The box is held in position for the investment period.
(3) When investment is complete, surplus mixture is emptied into the machine through the knock-out grid, and the core is transferred to the separate oven for curing.

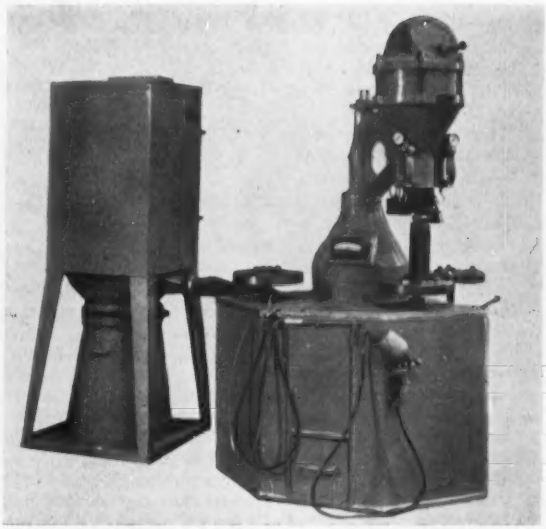


Fig. 4. Polygram bench type shell core maker Mark IX

Mould closing

For two reasons the use of mechanical devices, such as clips and bolts, for closing a shell mould is far from satisfactory. In the first place, their action is of necessity localized and cannot be relied upon to exert uniform pressure over the complete surface of the mould, and a uniform pressure is highly desirable. Secondly, the time taken to close a mould by mechanical devices will usually be unduly long in relation to the other time cycles in the complete process.

Hot setting adhesives are used in the Polygram method of assembling shells, and two pieces of equipment have been developed for closing shell moulds rapidly and effectively. They are a mould heating oven, shown at the left in Fig. 6, and a shell mould closer, shown at the right in Fig. 6. Adhesive is applied to one half of a cold shell mould, and the heat required to set it is supplied by the other half mould, which has been brought to the correct temperature in the mould heating oven.

The temperature to which the top shell is raised is critical and the Polygram mould heating oven has been designed to maintain this temperature to within ±10 deg C irrespective of the period that the shell remains in the oven. It is also so designed that once a shell has been placed in the oven it cannot be removed until it has reached the operational temperature. The

Fig. 5. Polygram-Croning shell core maker Mark XI

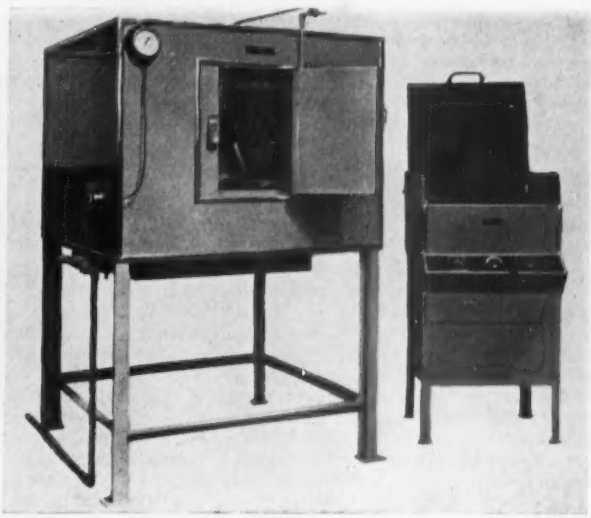


Fig. 6. Polygram mould heating oven (left) and Polygram shell mould closer

special features of this oven are:—

(1) Opening the oven door automatically rotates the cold shell into the heating zone and presents a fully heated shell to the front for removal.
(2) The oven does the duty of an automatic tunnel stove at a very much lower initial cost.
(3) It is gas-fired and consumes only 45 ft^3 per hour.
(4) There is automatic temperature control.

An extremely ingenious method has been devised for exerting a uniform pressure over the whole surface of the mould during the closing operation. In the Polygram closer the mould is completely enclosed between two rubber diaphragms, of which the upper one is mounted on a hinged frame to permit the mould to be inserted. Air is then exhausted from the space between the diaphragms and a uniform pressure of approximately 15 lb/in^2 is exerted on both sides of the mould irrespective of its contours. This pressure is maintained for a sufficient period to allow the adhesive to set, after the top diaphragm is hinged up and the mould, ready for casting, is removed. This unit will give an output of about 60 moulds per hour.

An intimate mixture of sand and resin is an essential factor in successful shell moulding. For this reason, the Company have developed a mixer which, in addition to being efficient, is cheap, simple to operate and is dust proof. It consists essentially of a frame that carries two parallel shafts which are driven simultaneously by an electric motor through a vee rope reduction train. These shafts carry four rubberized rollers so spaced as to engage on and rotate the mixing drum. A special removable agitator assembly is fitted inside the drum, which is closed at one end by a lid with a rubber gasket. Two drums are supplied with each machine.

For successful shell moulding it is essential to use special sands that are much more expensive than those used in conventional sand moulding. Hitherto, it has been the practice of Polygram Casting Company Limited to use new sands of the finest physical and chemical characteristics in the making of cores and moulds. This practice reduced the number of variables by establishing a material constant. In the interests of economy, investigations have been carried out into the qualities of reclaimed sand. They have been reassuring in that they show that the material constant is not greatly altered and the Company have now adopted the practice of reclaiming sand. For this purpose they have developed a simple, but effective, sand reclamation oven. It is operated as follows:—

(1) A gas torch is applied to an aperture in the base.
(2) The oven is filled with broken moulds to a depth of about 12 in.
(3) When, in about 10 minutes, the mould pieces are ignited, the gas torch can be withdrawn.
(4) Complete filling the oven. Reclaimed sand will trickle through the base to a suitable container.

This unit has the following special features: other than labour for loading, there are no running costs; after initial ignition has occurred, combustion is self-supporting and reclamation is continuous so long as moulds are loaded into the oven; sand is reclaimed at the rate of 1-1½ cwt per hour.

AIR- OR WATER-COOLING

A PAPER entitled "Air- or Water-Cooling" by R. Kloss has been published in *A.T.Z.*, November, 1953. This paper is a reply to another on the same subject, by W. D. Bensinger.

The author states that water-cooling systems in high-speed engines have been improved in recent years, notably by the use of quick-response thermostats and pressurized cooling. These improvements have reduced engine sensitivity to unfavourable operating conditions and to sulphur-containing fuels. They have also increased engine life to an economically tolerable level. Further increase in coolant temperature has been precluded by the physical properties of water. Advantages of air-cooling include operational safety, economy, and insignificant maintenance requirements. Nevertheless, water-cooling is obviously preferable for vehicles where silent running is a prime requirement.

Performance data show that air-cooled engines can very well compare with the Daimler-Benz engines mentioned by Bensinger, provided swept volume, number of cylinders and rotational speeds are similar. For the same swept volume, the air-cooled engine is 23 per cent lighter, though the greater heating of the intake air reduces the output by 6 per cent. However, the weight per horse power is 17 per cent less, and the advantage in weight and length would not be lost by increasing the swept volume by the 6 per cent necessary to obtain equality of output.

With reference to the desirability of high specific outputs, it is stated that even in aircraft engines, air-cooling is preferred In the Volkswagen, the relatively low specific output is due to a deliberate design policy, and not to the air-cooling system employed. Again, as to oil-coolers, the author's view is that linking the oil-cooler to the water-cooling system is a mistake, since the warming-up period is prolonged with the engine cold, and heat-transfer from the oil is poor when the engine is hot, because of the small temperature difference. With air-cooled oil-coolers, the air intake is closed during the warming-up period, while under full load there is a considerable temperature difference between the lubricant and the fresh, cold air. These and further considerations, including production figures for both types of engines, lead to the conclusion that the real point at issue is when to use air-cooling and when to use water-cooling. *M.I.R.A. Abstract No.* 6760.

TYRE PERFORMANCE ON ICE

AN article entitled "Study of the Factors Affecting the Friction of Tread Compounds on Ice," by C. S. Wilkindon, Jr., appeared in *India Rubber World*, July 1953. Experiments to determine the frictional properties of rubber on ice have generally been made with motor vehicles on frozen lakes, but this article describes laboratory tests made on small rubber specimens in an apparatus contained within a refrigeration chamber.

Various factors affecting friction, including the velocity of sliding, type of path and method of applying the load, were studied. With increase in velocity from zero to 100 cm/sec, the coefficient of friction 'k' first rose rapidly and then fell gradually. This was clearly shown on a graph plotted to log scale; and 0·8, the maximum value of 'k,' was reached at −20 deg C. At −10 deg C it was about 0·5. Other curves showed that 'k' fell progressively with increase in load. This effect was attributed more to local melting of the ice owing to frictional heat than to the lowering of the melting point by pressure. There appeared to be little difference in frictional load whether the three specimens ran on one single path, or on three separate ones. As temperature was reduced, 'k' reached a maximum value of about 1·2 at −30 deg C with a 6 lb load, but maxima could not be reached at the higher loads, owing to frictional heat developed. By increasing the pressure from about 40 to 200 lb/in^2 at constant temperature, 'k' was reduced from about 0·2 to 0·1. *M.I.R.A. Abstract No.* 6654.

ENGINE STALLING

A Study of Diesel Behaviour

By P. H. Schweitzer

A STUDY of the stalling behaviour of engines in general and of diesel engines in particular seems overdue in view of the difficulties experienced with diesel engines under severe load conditions, such as earth moving, rough terrain hauling and slow idling. All users want engines that can "lug" and that can idle slowly without danger of stalling, but before stall-proof engines can be designed more must be known about stalling behaviour and the factors that influence it.

One type of stalling may be dispensed at the outset. It is a simple fact that whenever the horse-power needed to carry the load is greater than the maximum horse-power of the engine, stalling will occur, either at once or as soon as the inertia stored in the flywheel and other masses is dissipated. This type of stalling has no part in the following discussion. However, engine stalling is often experienced when the engine has the power to carry the load. That means that equilibrium between the power output and power input of the machine is possible. It is not the lack of equilibrium that causes the engine to stall, but the lack of stable equilibrium, as will be seen from the following discussion.

Mechanics of stalling

Consider a prime mover with a rotating shaft, the engine, coupled to a machine, which may be any mechanism that turns when the necessary effort is applied to it. The machine may be a vehicle, a saw, a pump, a propeller, or one-half of a hydraulic torque converter — the half that is coupled to the engine. When running, the engine develops torque or turning effort and the machine absorbs torque, and at a constant rotational speed an equality exists between engine torque, T_e, and machine torque, T_m, so that $T_e = T_m$.

If at any one time in a coupled system $T_e > T_m$, the rotative speed will no longer be uniform. The surplus torque $T_e - T_m$ will accelerate the system until it either reaches equilibrium again or wrecks itself. On the other hand, if at any one time in a coupled system $T_e < T_m$, the system will begin to lose speed and will continue to lose speed until it either reaches equilibrium at another speed or stalls.

This is elementary until the speed comes in. Every engine has a definite torque characteristic; that means it develops a certain amount of torque, lb-ft, at a certain r.p.m. The torque: r.p.m. curve of an engine may refer to a given fuel control setting or to a given governor setting, to a given accelerator pedal setting, or to another given setting arbitrarily specified. Every machine also has a definite torque characteristic, which means it will absorb a definite amount of torque, lb-ft, at a certain shaft speed for a given setting.

A typical torque curve of a carburettor engine is shown in Fig. 1a. It drops considerably from the maximum torque to the maximum speed torque even at full throttle and more at part throttle. The speed corresponding to this maximum torque is lower than the speed at which maximum power is developed. A typical engine torque curve referring to a diesel engine with a fixed fuel control-rod setting is shown in Fig. 1b. The drop in the torque curve between maximum torque and maximum horse-power is less.

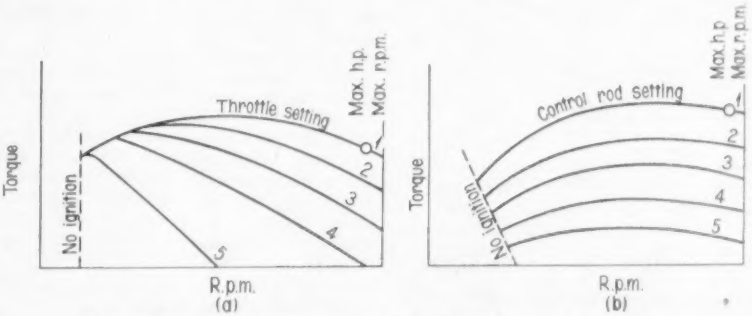


Fig. 1. Engine torque curves. (a) carburettor engine. (b) diesel engine

The torque characteristics of power absorbing machines vary greatly. Three typical curves are shown in Fig. 2. When an engine coupled to a machine runs at a common uniform speed, equality exists between the engine output torque and machine input torque determined by the point of intersection of the two torque curves, as shown in point A in Fig. 2. A diesel engine coupled to a lift truck shows also another point of intersection at B. A road vehicle having a torque curve like the dashed line also has another intersection at C, but a propeller curve would have not more than one intersection. Constant speed operation can only take place at points of intersection.

When two points of intersection exist, stable operation can only take place at A. At B or C the operation is unstable, which means that in practice it cannot take place at all except for a very short period. Now visualize equilibrium operation and consider the effect of a minute disturbance. Such a disturbance can be caused by a slight variation of the load or by a cycle-to-cycle variation of the fuel feed, or one cylinder missing in one cycle, or for other reasons. Suppose that while the unit is operating at A a little less fuel is introduced during *one* cycle which causes a minute speed drop from n to n^1. This will cause the engine torque to increase from T to T^1. At this point there will be a surplus engine torque which will cause the couple to speed up. Equilibrium will be quickly restored at A^1. We call $T_r = T^1 - T$ = restoring torque. The restoring torque is positive at point A when the engine loses speed and negative when it gains speed and so it tends to prevent the engine from stalling or racing.

If the engine operates at point B or C and a disturbance occurs, the result will be different. A drop in speed, caused, for instance, by a

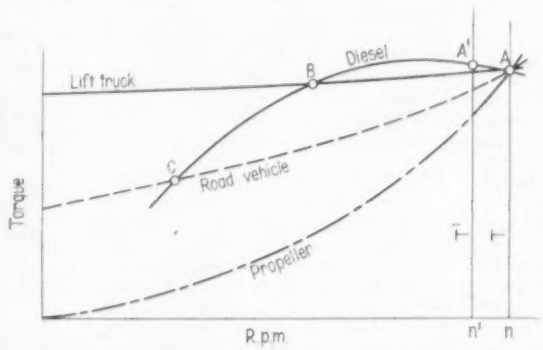


Fig. 2. Torque curves for various machines coupled to a diesel engine

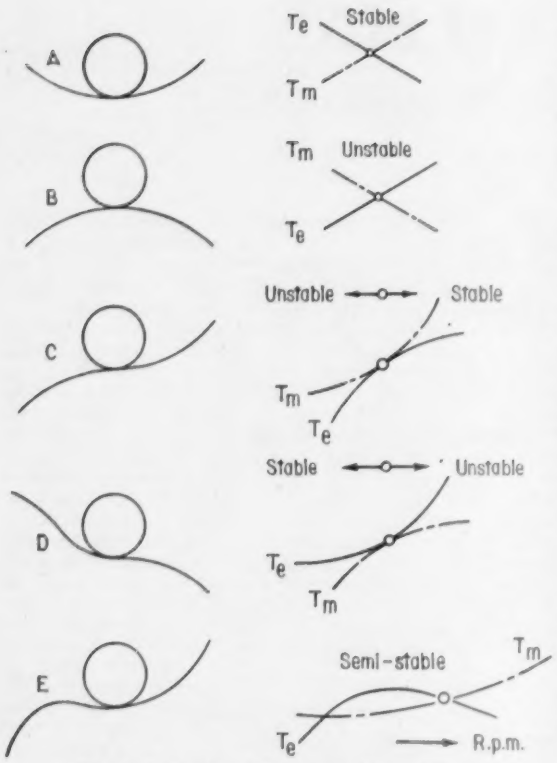


Fig. 3. Ball analogy of engine-machine stability

little less fuel being fed during a single cycle, will produce a negative restoring torque. This will cause a further speed drop and with it a greater deficit in torque. This disequilibrium grows from cycle to cycle until the engine stalls. Since such minute disturbances are unavoidable, an engine-machine couple is not able to operate at intersection B or C. Only a stable equilibrium permits practical engine operation. A stable equilibrium is characterized as where:

1. Engine torque = Machine torque
2. Engine speed = Machine speed
3. There is positive restoring torque for slight speed drop.

Expressed mathematically the condition of stable equilibrium at n_o r.p.m. is:

$$\frac{T_e = T_m}{n} \quad \ldots\ldots\ldots\ldots (1)$$

$$\frac{d(T_e - T_m)}{dn} < 0 \quad \ldots\ldots (2)$$

In what follows, the negative value of the derivative $d(T_e - T_m)/dn$ is termed Slope of Restoring Torque, abbreviated SRT. For an engine to run stable the SRT must be positive, that is, SRT>0. If SRT is for any reason negative, the engine-machine couple is not in stable equilibrium and will stall or race.

These relations can be visualized by envisaging the stability of a ball on a surface. For the ball to be in equilibrium, the resultant of all forces acting upon it must be zero. This happens in positions A, B, C, D and E in Fig. 3. Position A represents stable equilibrium because a minute displacement right or left will produce a force which compels the ball to return to its original position. The ball can also be in equilibrium in position B, but that is an unstable equilibrium because a minute displacement will force it to run away. In position C the ball is *stable* against an infinitesimal displacement to the right but unstable to same to the left. In position D it is the other way. In position E the ball is stable, but not very, because a small displacement to the left will upset it. The corresponding torque curve intersections are shown on the right where the engine and machine torques are indicated near the intersection point against r.p.m. increasing to the right.

As seen from position E, the fulfilment of stable operation is not sufficient to insure stable operation in practice. The angle of intersection also has an influence. A wide angle intersection as shown by the propeller curve, Fig. 2, is best. The intersection shown by the road vehicle at A is fair. A very acute intersection as shown by the lift truck is unsatisfactory because then a small increase in load torque (or a small decrease in engine torque) makes a large decrease in r.p.m., and the equilibrium point may slide from A to B, which surely produces stalling.

This makes it plain that the stalling characteristic of an engine depends as much on the coupled machine as on the engine itself. Machines which by nature have rapidly rising torque characteristics (like a propeller) are difficult to stall because, no matter how unfavourable the engine torque curve is, they rarely allow an unstable equilibrium. Shovels, on the other hand, stall easily because often the restoring torque is small or negative.

Factors influencing engine stability

Let us now turn our attention to the engine. Obviously, the more declining the engine torque curve is, the more stable the engine is and the less likely it will stall if coupled to a machine. While the torque curves of prime movers display a great variety (consider the steam engine on one hand and the synchronous electric motor on the other hand), the torque curves of internal combustion engines are pretty much alike and look like a hill with gentle slopes to the right and to the left of the peak. At any one speed the torque is determined by the amount of fuel burned per revolution and by the thermal efficiency of the conversion of the fuel energy into mechanical work.

If the specific fuel consumption is known for any speed-load combination, from the equation:

$$\frac{F}{f} = \text{Horsepower} = \frac{2\pi T_e n}{33{,}000} = \frac{60nGz}{f}$$

the torque is obtained as:

$$T_e = \frac{33{,}000 \times 60 \times z}{2\pi} \cdot \frac{G}{f},$$

where T_e = engine torque, n = r.p.m., F = total fuel consumption, lb/hr, f = brake specific fuel consumption, lb/b.h.p-hr, G = fuel fed per revolution per cylinder, lb, z = the number of cylinders.

It is seen that if G/f is independent of the speed, T_e is constant and the torque curve becomes a horizontal line. In the case of a naturally aspirated carburettor engine, the amount of fuel introduced per cycle bears a more or less constant relation to the amount of air aspirated, which is equal to the engine displacement multiplied by the volumetric efficiency. Therefore, for any throttle setting:

$$T_e = \text{Const} \frac{\eta \text{ vol}}{f}.$$

If the volumetric efficiency decreases with speed as the top curve and the specific fuel consumption varies as the bottom curve, the torque must vary as shown in Fig. 4.

With a diesel engine the amount of fuel introduced per cycle is not controlled by the amount of air taken in but is controlled independently by the fuel control rod of the injection pump. For a given control rod setting the injected quantity varies with the pump speed. In Fig. 5, the specific fuel consumption and the injected fuel quantity per cycle are shown for a diesel engine and the torque curve calculated from the above equation. For another control rod setting a different torque curve is obtained. A deciding factor in the shape of the torque curve of a diesel engine is the variation of the injected fuel quantity per cycle with the pump speed.

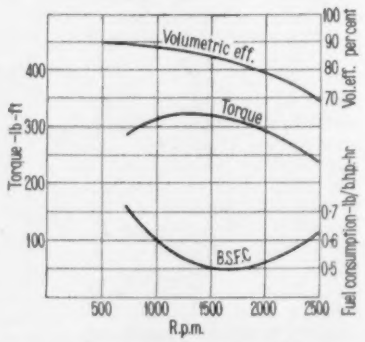


Fig. 4. Calculated torque curve for a carburettor engine. Air-fuel ratio 12:1, inlet air density 0·0685 lb./ft³

The popular port-controlled injection pump has a rising delivery characteristic which means that the injected fuel quantity (usually expressed as mm^3 per cycle) increases as the pump speed increases. The main reason for this behaviour is the so-called "afterflow." Delivery does not cease at the instant the plunger helix uncovers the spill port but continues for a while afterwards. This afterflow is greater at high pump speeds and produces a rising delivery characteristic.

If the torque curves of the carburettor and diesel engines are compared, from the standpoint of stability the comparison is unfavourable to the diesel which peaks at a relatively higher r.p.m. but for good economy is run mostly at a low r.p.m. Furthermore, the maximum torque of a diesel engine is relatively lower than that of an Otto engine. The "elasticity" of an engine is expressed by Floessel[1] by the index:

$$E = \frac{T_{max}}{T_{at\ rated\ h.p.}} \cdot \frac{n_{at\ rated\ h.p.}}{n_{at\ max\ torque}}$$

elasticity is, however, an artificial index at best. The stalling characteristics of an engine can best be examined by observing or calculating how the engine behaves under service load. A graphical analysis similar to that shown in Fig. 2 is a convenient method to determine stalling characteristics.

In Fig. 6 upon the torque curves of a carburettor engine are superimposed the torque curves of a vehicle at high gear. The engine torque curves fall steeply at part throttle. The reason

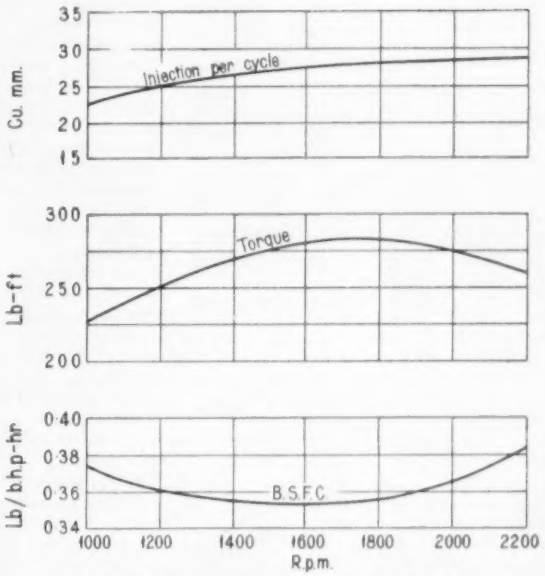


Fig. 5. Calculated torque curve of a diesel engine

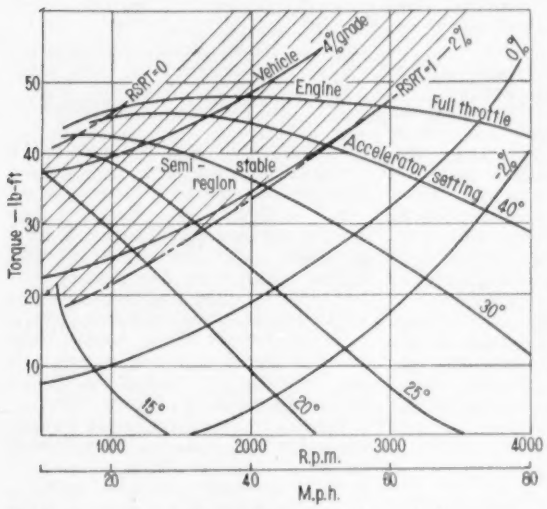


Fig. 6. Torque curves of a carburettor engine and of a passenger vehicle at high gear

for this is that the air flow past the butterfly valve diminishes rapidly with speed when the butterfly is partly closed. This is responsible for the stable characteristics of the carburettor engines. The vehicle resistance torques are shown for level road, 2 per cent, 4 per cent and −2 per cent grade. The torque curves are merely displaced vertically by the amount required to compensate for the force required to overcome the gravity.

cation. The points corresponding to SRT=0 are those where the vehicle torque curves are tangent to the engine torque curves. All torque speed combinations to the left of the RSRT=0 line are unstable and liable to cause stalling, as will be explained below.

That the SRT is positive is no sure guarantee, however, that the engine will not stall. It only means that an infinitesimally small disturbance will not stall the engine. To be reasonably stallproof, the SRT must have a certain magnitude. How much depends on circumstances.

In order to express the torque relationship in a non-dimensional manner and to make it independent of the scale or units used, the concept "Relative Slope of Restoring Torque"

The elasticity of an engine is somewhat indicative of its resistance to stalling. From Figs. 4 and 5, the elasticity of the Otto engine is $\frac{325}{240}\cdot\frac{2500}{1400}$ = 2·42, and that of the diesel engine $\frac{267}{225}\cdot\frac{2200}{1750}$ = 1·5. From Floessel's statistics the elasticity of most four-stroke cycle Otto engines varies from 1·5 to 4·5 with an average of approximately 2·4. That of the two-stroke cycle Otto engines is somewhat lower. That of the four-stroke cycle (automotive) diesel engines varies between 1·2 and 2·0 with an average of 1·79, and that of the two-stroke cycle diesel engines between 1·44 and 2·79 and averages 2·45.

Contrary to popular belief the diesel is less "elastic" than the gasoline engine. What some people may have had in mind in describing the diesel "elastic" is that it responds more readily to the accelerator and has, therefore, a better pick-up. Floessel's

At any grade continuous operation is only possible at the intersection points of the engine torque curves with the vehicle torque curves. But not each intersection represents stable operation. As stated above the slope of the restoring torque must be positive.

$$SRT = -\frac{d(T_e - T_v)}{dn} > 0$$

otherwise the engine will stall at the slightest provo-

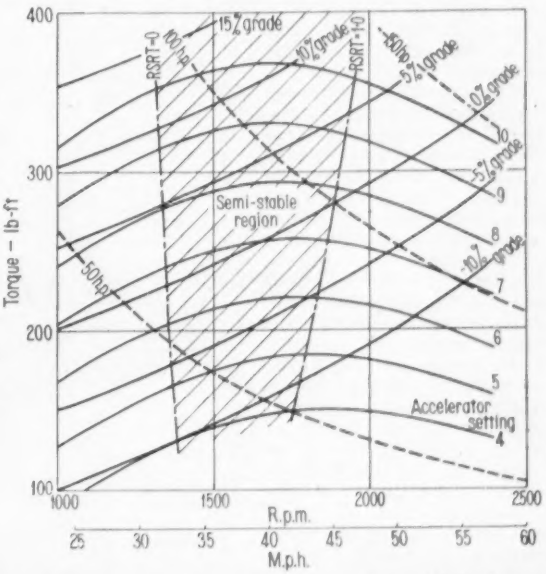


Fig. 7. Torque curves for a diesel engine and a truck at high gear

(RSRT) is introduced, defined as:

$$RSRT = -\frac{d(T_e - T_m)}{dn} \cdot \frac{n}{T}$$

and can be visualized as the per cent increase of restoring torque divided by the per cent increase in r.p.m. over a small increment. In these equations RSRT is independent of the scale of plotting the torque curves and of the units used. RSRT can be regarded as an index of stability; the greater it is the more stable the engine-machine couple is and the less likely to stall.

Aside from the torque relationship,

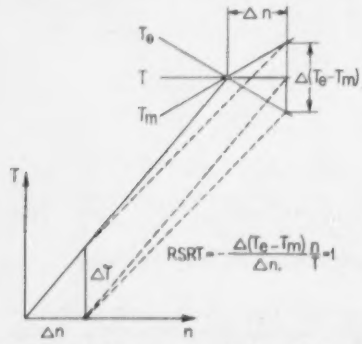


Fig. 8. Graphical construction for the slope of restoring torque in an engine-machine couple in the case of semi-stable operation

the stalling tendency of an engine-machine couple depends largely on the following factors:

1. Magnitude of transient disturbance.
2. Inertia of engine and coupled machine.
3. Governor.

If $SRT = -\frac{d(T_e - T_m)}{dn} < 0$ and the inertia of the coupled system is zero, the smallest disturbance, such as the slightest reduction of fuel during one cycle or the slightest momentary increase in machine resistance will stall the engine. The greater the transient disturbance the more likely the machine is to stall.

The greater the inertia, the less likely the machine is to stall. In figuring inertia, to the flywheel inertia, the equivalent inertia of the engine and the machine (vehicle) must be added. Therefore, a heavier vehicle is less likely to stall.

A governor is an artificial anti-stalling device. No engine can stall without losing speed. If speed loss is prevented by the governor the engine is stallproof. The reason governors do not always prevent stalling is that they lack the capacity or the speed of response of governor or engine to neutralize instantaneous disturbances in the torque equilibrium. But they are always helpful, especially if the RSRT is small.

To make a vehicle reasonably stallproof, RSRT must exceed a certain value. We may select an arbitrary value of 1·0 and call it the line of stable operation. When RSRT = 1, 1 per cent speed drop produces 1 per cent torque rise. Points corresponding to RSRT < 0 belong to the region of unstable operation where stalling is difficult to avoid. The region between 0 < RSRT < 1 may be called the region of semi-stable or quasi-stable operation.

Reverting to Fig. 6, the line corresponding to RSRT = 1 has been plotted and it falls to the right of the RSRT = 0 line. It almost coincides with the vehicle torque curve for 2 per cent grade. Below and to the right of RSRT = 1 line the engine-machine couple is termed stable. The region between the two RSRT lines is termed semi-stable and above the RSRT = 0 line unstable. It is seen that this gasoline engine mounted in a suitable vehicle produces stable operation in the practical region. When the grade is such that operation falls in the semi-stable region, gear shifting is called for. This happens to coincide closely with the 2 per cent grade.

Next, subject a diesel engine mounted in a vehicle to a similar analysis. The engine is a Krauss-Maffei four-cylinder $4\frac{7}{8}$ in × $5\frac{1}{2}$ in two-stroke cycle diesel engine (described by Sass and Schweitzer in *Automobile Engineer*, July 1952) and the vehicle a 15 ton truck with a gear transmission which gives one mile per hour at 41·7 r.p.m. at high gear.

In Fig. 7 the engine and vehicle torque curves are plotted for various fuel control rod settings and various grades. Where the vehicle torque curves are tangent to the engine torque curves RSRT = 0. This is the instability line. RSRT = 1 line is also established, where the intersections are of such angle that:

$$RSRT = -\frac{d(T_e - T_m)}{dn} \cdot \frac{n}{T} = 1{\cdot}0$$

for which the graphical construction shown in Fig. 8 can be used. In this way the field is divided into stability, semi-stability and instability regions.

First, we note that the region of instability is very large, almost all below 32 miles per hour, irrespective of grade. The region of semi-stability extends up to approximately 45 miles per hour and only above that is the vehicle

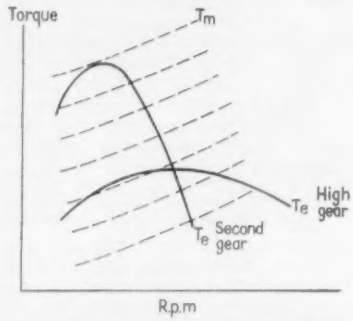


Fig. 9. Effect of gear ratio on stalling

operation "stable." Somewhere between 32 and 45 miles per hour it is advisable to shift to a lower gear to avoid stalling. This analysis again ignores the governor action. With a governor, stable operation is possible below these speeds. But a comparison between Fig. 7 and Fig. 6 clearly shows how much less stable the diesel engine is than the carburettor engine, both being mounted in a road vehicle. It becomes very clear why diesel engines cannot get along without governors.

Knowing what makes it stall gives us clues how to make an engine more stallproof. We can quickly dispose of the very obvious ways to achieve this purpose in order to concentrate on those that are less obvious. Making the engine bigger or the load smaller obviously pushes the intersection points B and C in Fig. 2 to the left and reduces the stalling r.p.m. A similar effect can be obtained by making the load more steeply rising with the speed (propeller, pump, etc.).

The most popular way to make an engine-machine couple more stallproof is to increase the transmission gear ratio. What a doubling of the gear ratio between engine and the rear axle does to a vehicle is seen from Fig. 9. The stalling torque increases to almost double and the stalling r.p.m. becomes substantially lower. Hydraulic and electrical transmissions have similar effect.

Restricting consideration to the engine itself, the most effective way to improve stability is by proper control

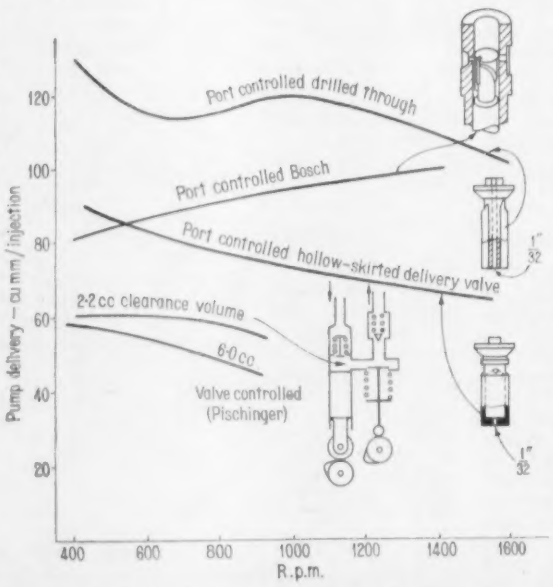


Fig. 10. Speed-delivery characteristics of injection pumps

of the fuel introduction. It was noted that the carburettor introduces the fuel in such a way that the fuel feed per cycle decreases rapidly with the r.p.m. This makes the carburettor engine inherently stable.

Speed-Delivery Characteristics of Injection Pumps

In diesel engines the fuel injected per cycle varies less with the r.p.m., sometimes it slightly decreases, more often it slightly increases. Fig. 10 shows some typical speed-delivery characteristics of diesel injection pumps.[2,3,4] It is fairly well known that the speed-delivery characteristic of the widely-used port-controlled jerk pumps is a rising one, which does not favour stability. The geometric displacement of the plunger does not vary with the pump speed, only with the rack setting. Therefore, if the actual delivery were equal to the geometric delivery, it would be represented by a straight line, parallel to the r.p.m. axis. The actual volumetric delivery of a jerk pump per pump cycle can be represented as:

D actual = D geometric − Incomplete filling − Compressibility − Leakage − Backflow − Retraction + Preflow + Afterflow.

D geometric is equal to the plunger cross-sectional area times effective stroke, from inlet port closure to spill port opening. In a given pump this is controlled solely by the control rod or rack position.

Incomplete filling is the result of throttling through the inlet port and is equal to the volumetric amount of vapour and entrained air in the fuel at the time of inlet port closure. These are determined by the vapour pressure of and the entrained and dissolved air in the fuel and the absolute pressure in the barrel at inlet port closure. High supply pressure, large inlet port and slow pump speed improve the filling; entrained air, dissolved air and high Reed vapour pressure have the opposite effect.

Compressibility of the fuel is approximately 3 per cent per 1,000 lb/in^2 compression. The higher the peak barrel pressure and the larger the pump clearance volume the greater the compressibility.

Leakage is the volumetric amount of fuel which escapes from the system because of plunger clearance, injector-needle clearance, and imperfect seals in the pump, line and injector.

Backflow is the volumetric amount of fuel which flows back from the line to the pump barrel past the delivery valve shortly after the opening of the spill port. A delay, due to inertia and friction, in the closing of the delivery valve results in backflow and is greater with softer delivery valve spring.

Retraction is incorporated in many delivery valves. When a retracting delivery valve closes shortly after cut-off, the receding valve acts as a piston and sucks a certain amount of oil back from the line. Its purpose is to relieve the line from residual pressure and so prevent dribble and secondary openings of the injector needle.

FACTORS IN DELIVERY CHARACTERISTICS

	Increases Delivery	Absolute Magnitude Increases With Pump Speed	Effect on Engine Stability
Geometric Delivery	+	0	Neutral
Incomplete Filling	−	+	Favourable
Compressibility	−	+	Favourable
Leakage	−	+(1)	Favourable(1)
Backflow	−	+	Favourable
Retraction	−	?	Insignificant
Preflow	+	+(2)	Adverse(2)
Afterflow	+	+	Adverse

(1) It depends on the line resistance and valve opening pressure. If these are small, leakage decreases with speed because of less time available; if they are large, leakage increases with speed because of increased barrel pressure.
(2) Probably insignificant.

Preflow is the volumetric amount of fuel delivered before the inlet port is closed. The throttling in the inlet port just before its closure is sufficiently high to force some fuel into the line during this period. In our tests[5] preflow was insignificant.

Afterflow is the volumetric amount of fuel delivered after the spill port opens, due to liquid inertia and throttling through the spill port. In our tests[5] afterflow was considerable and increased steeply with pump speed.

For declining delivery characteristic the positive members in the equation should be either small or they should decline with the speed. The negative members should increase with speed. Their general behaviour is shown in the accompanying table. In order to improve engine stability, the factors having favourable effect should be emphasized and the factors having adverse effects minimized or neutralized.

Incomplete filling could be emphasized by reducing the inlet port. A disadvantage is that equality in metering among several cylinders is then more difficult to achieve. In single-plunger pumps, however, "strangulation metering" is practical and in use.

Compressibility is emphasized by increasing the clearance volume of the pump (see low curves in Fig. 10). A disadvantage is greater tendency to airlock and greater cycle-to-cycle variation in injection timing. Complete elimination of the delivery valve produces the same effect only more so.

Leakage is favourable if the valve opening pressure and/or the line resistance is fairly high. For this reason leakage can be artificially increased by a small hole through the valve as shown on the top curve of Fig. 10. Leakage is more effective if combined with a restriction to the flow as shown by the hollow-skirted delivery valve, shown on Fig. 10. With that arrangement, nicely tailored speed-delivery characteristics are produced.

Backflow caused by the delayed closing of the delivery valve favours engine stability. Backflow can be emphasized by using a soft delivery valve spring. A potential disadvantage is delivery valve surge and spring breakage. This can be partly minimized by a delivery stop valve which limits its travel.

Retraction has an insignificant effect on the speed-delivery characteristics. However, by providing a small clearance between the retraction piston and the barrel the speed-delivery characteristic is improved.

Preflow as measured by the pump angle between geometric port closure and the first lift of the delivery valve was between 0 and −1 deg from 200 to 1,600 pump r.p.m. for a plain delivery valve, which is negligible. For a retracting delivery valve it was somewhat more but varied only little over the speed range. It probably can be ignored.

Afterflow is more responsible for the unfavourable speed-delivery characteristics of a jerk pump than any other factor. The afterflow

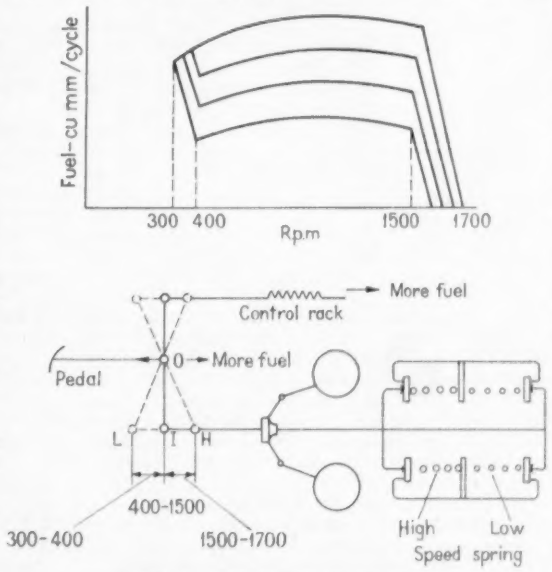


Fig. 11. Limiting-speed governor

increased from 1 deg at 200 r.p.m. to 24 deg at 1,600 r.p.m. with plain delivery valve, and from 9 deg to 44 deg with retractive delivery valve. Afterflow can be reduced by using larger spill ports of rhomboidal shape to produce faster spill but that is seldom done because of manufacturing difficulty.

As seen, the means to improve the speed-delivery characteristics of the conventional port-controlled jerk pump are rather limited. Nevertheless, distinct improvement can be achieved by modified delivery valves which, however, must be developed individually for the engine to be served.

Governors

The principle of a limiting-speed governor is seen from the schematic in Fig. 11. As long as the rotative speed varies between, say, 400 and 1,500 r.p.m., the flyballs occupy a constant position because the centrifugal force resulting from 1,500 r.p.m. is insufficient to compress the high-speed spring over its preload, nor is the opposite force resulting from a rotative speed of less than 400 r.p.m. sufficient to compress the preloaded low-speed spring. Consequently, the sleeve will remain in position I and the control rack will respond to the pedal pressure solely, I acting as a fulcrum for the lever.

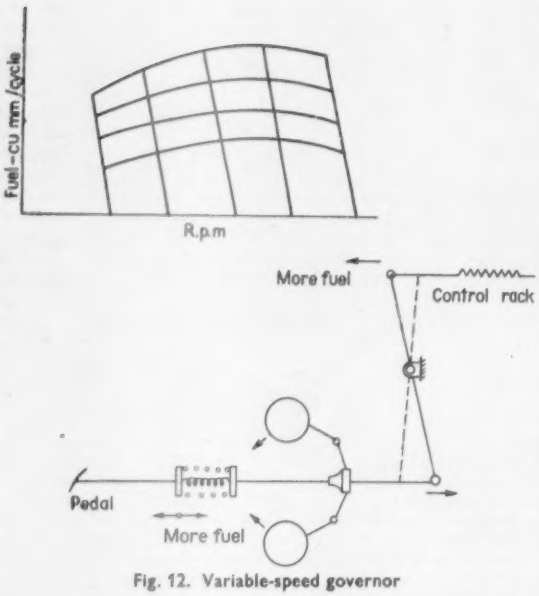
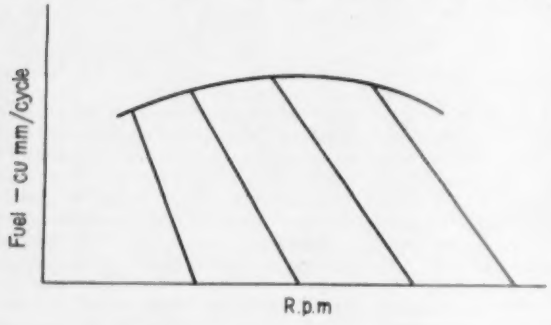
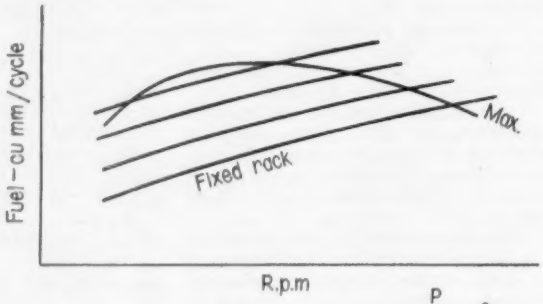


Fig. 12. Variable-speed governor

When the engine speed exceeds 1,500 r.p.m. the flyballs begin to move out compressing the high - speed spring, the sleeve moves toward H and turns the lever around fulcrum O, pulling the control rack to a reduced fuel quantity. This action prevents the engine going beyond 1,700 r.p.m. When the engine speed drops to 400 r.p.m., the flyballs begin to move in and the sleeve moves toward L while compressing the low-speed spring. This action pushes the control rack towards more fuel position and prevents the engine losing speed below 300 r.p.m.

In Fig. 11 (top) the ideal results of this type of governor are seen. If the





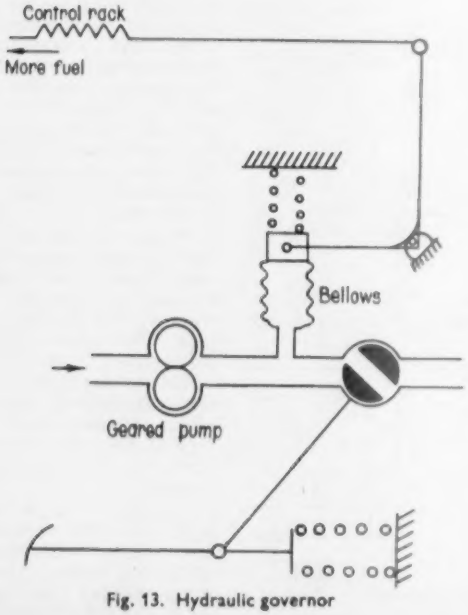


Fig. 13. Hydraulic governor

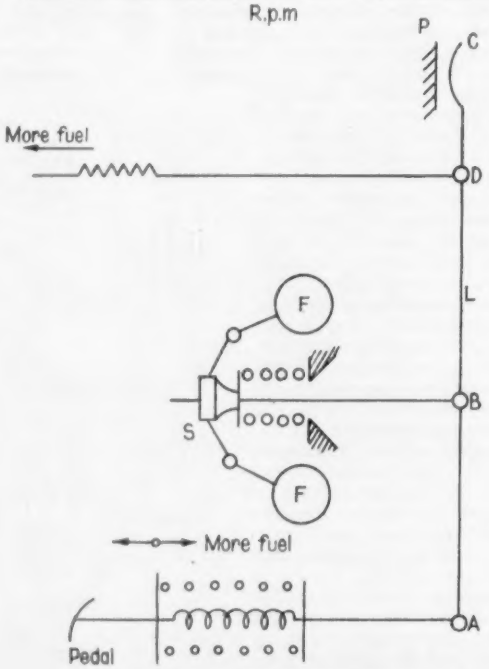


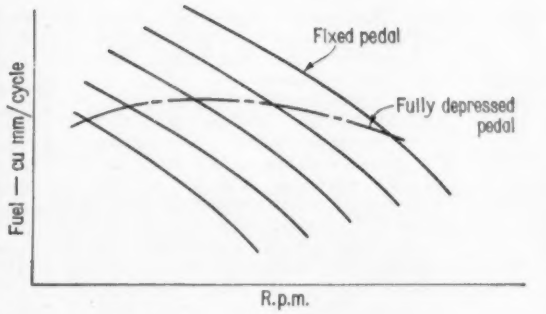
Fig. 14. Torque-control type governor

governor would act as described, the engine could never stall. Even though the governor does not influence the fuel quantity curve in the intermediate speed range, it provides a low limit to idling. Such a governor would effectively prevent stalling if the load disturbance were gentle and gradual. Actually, when torque curves are semi-stable, a more or less violent disturbance will make the unit slip into the instability region and then the engine will lose speed with such rapidity that the governor will be unable to stop it. Like a ball rolling down passes a small dip, the speed will roll down to complete stop.

Variable-speed governor.— Fig. 12 illustrates the principle of a variable - speed governor. When the driver depresses the pedal he compresses the spring, and the flyballs, acting against an increased force, move in. The engine gets more fuel. When the accelerator pedal is at rest, but the vehicle resistance increases, or the engine output torque decreases, and so the vehicle loses speed, the flyballs again move in, the linkages move in the direction of the arrows and the rack gives more fuel. The action of such a governor would result in a speed-delivery characteristic as shown on top of Fig. 12.

As pointed out by Coppens,[6] a disadvantage of this type of governor is its great bulk if it is made powerful enough to furnish adequate regulating force at low speeds. Another disadvantage is that it causes hunting, which is disagreeably felt particularly at low speeds. In addition, inasmuch as the control is exerted indirectly, through the governor spring, the response is more sluggish. If the governor is made more sensitive, in order to act quicker, its tendency to hunt is further increased.

Hydraulic governors. — Hydraulic and pneumatic governors have been used to correct the defects of the mechanical governors. When the hydraulics are used merely as a servomotor to increase the sensitivity of the governor, the difficulties are not completely eliminated. Some hydraulic and pneumatic governors do away with the flyball entirely and depend on the fluid pressure to actuate the sensing element. The principle of such a hydraulic governor is shown in Fig. 13 with the type of results it is expected to produce. The gear pump pumps fluid at a rate proportional to engine speed and its discharge pressure, communicated to the bellows, increases with the speed. When the speed drops the spring pressing the bellows deflects the linkage which pushes the control rack to more fuel. When the pedal is depressed more, the fluid discharge is less throttled, which has the same effect. Pneumatic and vacuum type governors employing the same principle are also in use.



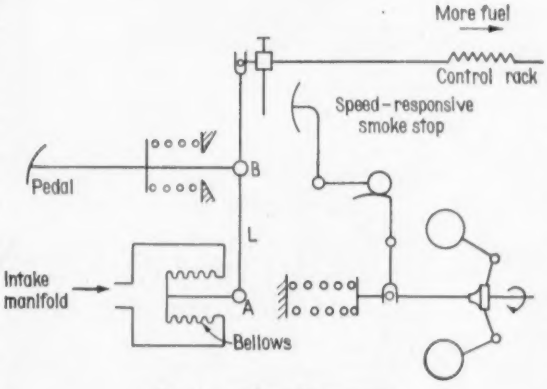


Fig. 15. Stallproof governor

Torque-control governors.—To improve the torque curve of automotive diesel engines, modified special torque-control type governors have been developed and are in use. Such a governor permits direct actuation of the fuel rack by the accelerator pedal, but with pedal fully depressed it automatically provides an increased amount of fuel when the speed decreases. The principle is shown in Fig. 14.

When the operator presses on the accelerator pedal, moving pivot A to right, lever L turns around fulcrum B, pushing pivot D to left and the control rack toward more fuel. When the accelerator pedal is fully depressed, cam C touches torque plate P and that limits the maximum fuel the control rack can deliver. But when the engine speed drops, flyballs F approach the axis and the spring forces the sleeve S to left, together with pivot B. This makes lever L turn round a point determined by the tangent of cam C and the torque plate P. Lever L can move thus because its lower pivot A is not solidly connected with the foot pedal but through the interposition of springs which allow some freedom for the lever to move while its upper end is fixed by the torque plate. When, due to the speed drop, the governor sleeve moves to left, pivot D also moves to left, pushing control rack to more fuel. Thus the speed drop is counteracted.

Under part load operation cam C is at a distance from torque plate P and inoperative. The engine obeys the foot pedal control, lever L turning around fulcrum B when the pedal position is changed. The fuel control obtainable with a torque-control governor is shown on top of Fig. 14. It is seen that the part load fuel quantity curves are unchanged, but superimposed on them is a maximum fuel curve which determines the torque under full load condition. This type of governor does not prevent stalling under part load.

"Stallproof" governor. — The ideal governor seems to be like the one depicted in Fig. 15. A bellows or diaphragm is in contact on one side with atmospheric air, on the other side with air under manifold pressure. In naturally aspirated four - stroke cycle engines we have partial vacuum there increasing with engine speed. In two-stroke cycle engines or supercharged four-stroke cycle engines with geared superchargers, the air box pressure is greater than atmospheric and increases with speed, roughly as the square of it. Fig. 15 refers to a two-stroke cycle engine.

While the engine speed is constant the bellows and fulcrum A occupies a certain position. When the operator presses on the pedal, lever L turns around fulcrum A, pushing the control rack towards more fuel. If the speed drops while the pedal position is unchanged, the air box pressure drops and the bellows expands to left, pulling lever L towards more fuel. The control rack ordinarily has a maximum fuel stop, frequently referred to as smoke stop. The fixed smoke stop can be replaced by a speed responsive smoke stop actuated by centrifugal force or other means. By a properly designed cam profile, to which the fuel control rack abuts, the selected maximum fuel curve or torque curve can be produced.

The kind of delivery characteristic possible with such a governor is shown on top of Fig. 15. The "natural" fuel curves are those shown in solid lines, declining steeply due to the square relation between air box pressure and engine speed. Even if the operator fails to react to an increase in load, the engine speed will drop but slightly because a speed drop automatically causes more fuel to be injected. There is, however, a maximum fuel for any given speed (determined by the maximum fuel cam) which comes into effect when the operator depresses the pedal fully, and which produces the full power torque corresponding to the fuel curve shown in dash-dotted lines superimposed on the "natural" fuel curves for given fixed pedal position.

Such a governor should prevent stalling not only at full load but also at part load. An engine equipped with

such a governor should be able to get by with fewer speed steps in the transmission and impose less severe demands upon the driver. The flyball actuated speed responsive smoke stop can be a relatively crude affair as a hunting of the flyballs will not cause hunting of the engine. The engine will respond to a change of pedal pressure almost instantly. Elements of all these governing systems can be found in the patent literature.

Stallproofing the load

It has been demonstrated that the stability of an engine-machine couple depends just as much on the characteristics of the machine as that of the engine. Changing the machine torque is, however, normally beyond our control. Nevertheless, in certain cases it is possible to improve the stability of the engine-machine unit by adding a fan, a blower, or a hydraulic device to the unit. Whether such a device is "useful" or parasitic, in either case it will have the effect of improving the stability of the unit and making the engine more stallproof.

Most engine-machine units have such components. The cooling fan, the lubricating oil pumps, the pump section of a hydraulic clutch or converter are anti-stall elements and so is the scavenging blower of a two-stroke cycle engine. The same can be said of the body of a fast vehicle having high air resistance. Streamlined bodies, on the other hand, make the vehicle less stallproof. In exceptional cases it may pay to hang parasitic loads of hydraulic or pneumatic nature on the unit and render it more stallproof thereby.

More effort has been expended in the last decades on improving transmissions than on improving the engines so that they may require less perfect transmissions. Neither approach should be neglected. The engine should be judged on how it behaves when connected to its real load, rather than to a dynamometer.

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RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

The Modern Diesel

By Donald H. Smith, A.M.I.Mech.E.

London: ILIFFE AND SONS, LTD., Dorset House, Stamford Street, S.E.1. 1954. 5½ in × 8¼ in. 256 pp. Price 12s. 6d.

Through successive editions since 1932, this handbook has provided up-to-date information on the characteristics of current high speed diesel engines and their fuel injection equipment. The latest edition, the twelfth, is the largest yet published. In its 256 pages, the development of the automotive type of diesel power unit, trends in fuel injection practice and combustion chamber design are discussed, and reference is made to fuel and lubricant developments.

Since the publication of the previous edition, there have been considerable developments in marine and industrial units founded on popular automotive types. This is reflected in the growing use of such engines in power craft, rail cars, portable electric generators, agricultural tractors and other industrial equipment. In the new edition, attention has been given to essentially automotive types, irrespective of their application to road transport vehicles. The engines of over 70 manufacturers from eleven countries are described.

The first four chapters, which describe the development and theory of the high-speed automotive diesel, are followed by a 31-page chapter on fuel injection systems and another of 26 pages on combustion chamber design. Chapter 8 comprises 71 pages in which are given the salient features and brief details of the majority of road vehicle diesel engines. The next four chapters are on railway service units, tractor and industrial engines, small air-cooled diesels and marine service units. The last chapter deals with diesels for private cars. Two appendices complete the work. The first gives, in tabular form, the leading particulars of all the British automotive type diesel engines in current production, and the second is a metric conversion table for bore, stroke and swept volume.

Gears

By H. E. Merritt, M.B.E., D.Sc.(Eng.) Lond., M.I.Mech.E.

London: SIR ISAAC PITMAN AND SONS, LTD., Pitman House, Kingsway, W.C.2. 1954. 5½ in × 8¼ in. 517 pp. Price 60s.

Dr. Merritt is so well known as an authority on gears as to need no introduction. Like its predecessors, the latest edition of this book is complete in scope and detail and is presented with clarity. It is a work which all engineers concerned with gears should have available, and which students will find invaluable.

This, the third edition of this well-known book, has been rearranged and largely rewritten and incorporates new material and new treatments. For example, separate chapters are now devoted to the geometry and detail dimensions of various types of gear, and a new chapter on some special types has been added. To make room for these features, material on gear trains has been deleted, but is contained in the companion volume, *Gear Trains*.

The rating methods given in earlier editions and in the relevant British Standards have, as a result of further study, been re-expressed in a way that leads to a more rational connection between the inevitable assumptions. This re-statement is accompanied by a full range of re-calculated charts. The author points out in his preface that despite the fact that these empirical rating methods are reasonably satisfactory, a true understanding of the mating behaviour of two surfaces and their lubricant still remains to be found

In addition to innumerable half-tone and line illustrations, there are 30 charts and 24 tables of data of considerable value to engineers concerned with the design and manufacture of gears. All the charts and most of the tables are given in appendices 1 and 2. A third appendix is a short but useful list of references. The work is completed by a further nine pages which contain an alphabetical index.

Designing by Photoelasticity

By R. B. Heywood, Ph.D., A.C.G.I., M.I.Mech.E.

London: CHAPMAN AND HALL LTD., 37, Essex Street, W.C.2. 1952. 5½ in × 8¼ in. 414 pp. Price 65s.

The author has had much experience of designing by photoelasticity and is now Principal Scientific Officer, Structures Department, Royal Aircraft Establishment, Farnborough. Formerly he was in the Research Laboratories of Rolls-Royce Ltd., Derby. His book is intended for designers as well as for photoelasticians. In this respect it differs from many others on photoelasticity and, as a result, would seem to the engineer to be of a more practical nature.

The necessity for designing parts which are light and yet of sufficient strength often makes it essential that the science of stress analysis should be applied. In this book, attention is drawn to the photoelastic method of stress analysis which, by recent simplifications in technique, has progressed from the field of pure research to that of applied research. The method needs to be applied by an engineer familiar with the design aspect of the problem rather than a physicist highly skilled in photoelastic techniques. A unique advantage of the photoelastic method is that it reveals, qualitatively or quantitatively as the case may call for, the modifications necessary to increase the strength of designs.

In this work the chapters are headed as follows: Behaviour of light in polariscope; Photoelastic materials; Models, their preparation and testing; Model stresses applied to prototypes; Frozen stress technique for three-dimensional analysis; Stress concentration in particular notches; Stress concentration in screw threads, bolts and nuts; Stress concentration at holes; Improvement of designs; The application of stress concentration factors. The last 44 pages of the book contain references and name and subject indexes.

NEW PLANT AND TOOLS

Some Recent Developments in Production Equipment

AN interesting version of their Model 2VR continuous milling machine has recently been produced by Adcock and Shipley Ltd., Ash Street, Leicester. This machine, which is illustrated in Fig. 1, has been specially developed for milling an automobile component in aluminium. The chief feature of the machine is a multi-station work table that is designed to allow work to be unloaded and loaded at some stations while milling is proceeding on work at other stations.

The standard machine consists of a conventional column with sliding head and a 26 in diameter, tee-slotted rotary table mounted on the knee. Both longitudinal and vertical adjustments are provided for the table. The sliding head has 3½ in vertical adjustment by means of a worm and worm wheel, which allows very accurate settings to be made. In addition, a micrometer stop is fitted, and a dial indicator can be incorporated for very fine setting. The machine spindle runs in precision, angular contact ball bearings. It is driven from a rotor stator unit built into the column, either directly through bevel gears, or through a back gear and then through a pair of bevel gears.

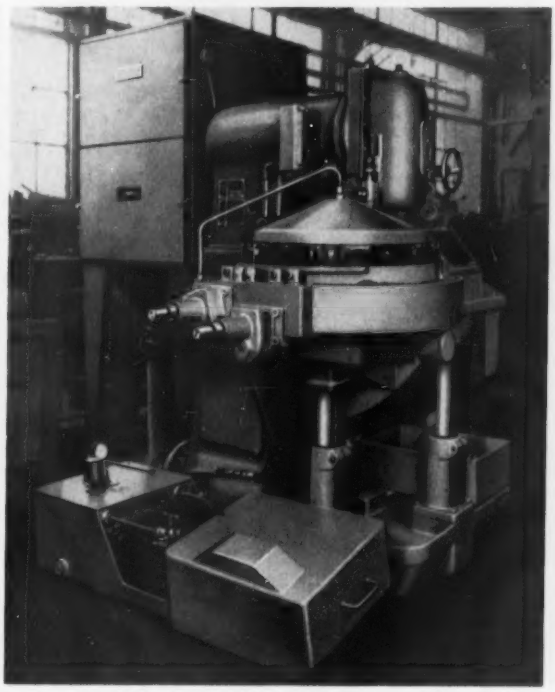


Fig. 1. Adcock and Shipley continuous milling machine

On the standard machine, the rotary table has eight tee slots on which work fixtures can be fastened. It is driven by a ¼ horse-power feed motor and there are nine different feed rates. On the special machine shown in the illustrations, the rotary table and the work-holding fixture are combined, and only longitudinal adjustment is provided for the withdrawal of the cutter from the work. The sliding head provides sufficient vertical adjustment for setting the cutters in relation to the work. Substantial locks are provided for all adjustments. On the standard machine an additional knee steady is provided, but on the machine illustrated two knee steadies are supplied to ensure greater rigidity.

To meet the special requirements for the straddle milling operation that is to be carried out, the standard table has been replaced by a special unit incorporating hydraulically-operated fixtures, for which a separate motor-driven hydraulic pump unit is mounted near the machine. The oil supply is fed under pressure to the centre of the fixture, and passes through radial valves to the radially disposed cylinders for operating the clamping mechanisms. One cylinder is used for each pair of fixtures.

As the table rotates, pressure is applied, first to one end of the cylinder to clamp the component, and then after the component has passed the cutters, oil is fed to the other side of the cylinder and the clamps are released. In the loading position, the component is lightly held by springs to retain it in the fixture until the full clamping pressure is hydraulically applied.

On this particular machine the main drive is by a single speed motor that develops 5 h.p. at 1460 r.p.m. This speed is reduced by a gearbox to a spindle speed of 658 r.p.m., which is equivalent to a surface cutting speed of 1000 ft per minute for the 6 in diameter cutters. Special high rake side and face cutters are employed. They give good surface finish even at the very high feed rates employed. The specified output of one component every 15 seconds is easily maintained.

Swarf disposal has been given special consideration. The fixture has been designed with a conically shaped top of sufficient steepness to prevent swarf from lying on it. In addition, rubber mouldings are used around the moving part of the fixtures to eliminate any possibility of small pieces of swarf entering them. A Dallow Lambert Dustmaster unit is incorporated to act as a swarf collector. It collects practically all the swarf into a suitable container at the rear of the machine.

Nut-runner

Hitherto the Ingersoll-Rand Impactool has been manufactured only in the U.S.A., but now the size 534 Impactool, illustrated in Fig. 2, is being produced in the Manchester works of Ingersoll-Rand and Co. Ltd., 165, Queen Victoria Street, London, E.C.4. This tool converts torque from a "multi-vane" motor running at a constant uniform speed into thousands of rapid rotary impacts. These impacts are transmitted to the socket, and nuts or cap screws can be tightened to any desired degree.

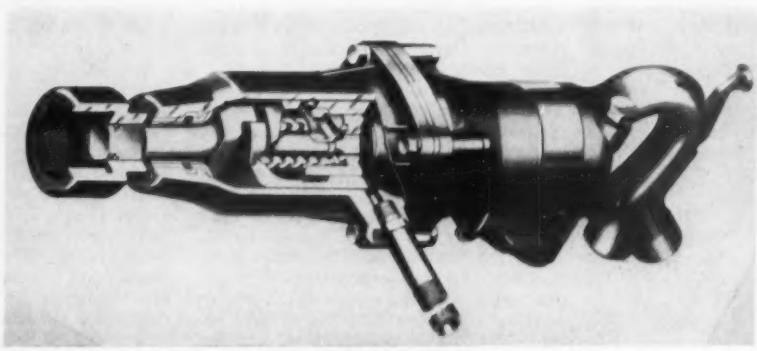


Fig. 2. Ingersoll Rand 534 Impactool

The direction of drive, or impacting, to tighten or to loosen a nut, is changed by reversing the direction of motor rotation. Reverse gears are not used. The motor drives the ball and cam type impact unit through a single heavy-duty planetary reduction gear, which is designed to take several times the load it must carry. There are only two external housings, both cast from aluminium alloys, heat treated for toughness and long life. Anti-friction bearings are used throughout, and a generous oil reservoir is built in to ensure a continuous metered supply of oil to the motor while the tool is running.

The operating principle of the patented impact unit does not produce sudden shocks or constant strains on the power end of the tool. This reduces the necessity for maintenance of the motor parts. Furthermore, the reaction on the operator is no greater when starting corroded-on nuts than when spinning loose ones down. This negligible torque reaction reduces operator fatigue and permits the tool to be used safely in any position. The tool applies 1,270 impacts per minute for tightening nuts to 1¼ in bolt size.

Fig. 3. Tilghman's Rotary-blast airless Wheelabrator

Airless Wheelabrator

An addition to the range of tumbling barrel type plants manufactured by Tilghman's Patent Sand Blast Co. Ltd., Broadheath, near Manchester, is illustrated in Fig. 3. It is designated the Rotary-blast, and is intended for use in plants where the volume of work does not warrant the installation of the larger sizes, which hitherto have formed the standard range.

The machine is airless in operation, and the effective cleaning action is provided by a 12 in diameter Wheelabrator unit, which is an integral part of the machine, and is situated at the end of the barrel opposite the door. This unit operates at a speed of 2900 r.p.m. and is directly coupled to a 10 b.h.p. high-torque motor. It should be mentioned that the Wheelabrator unit is exactly as fitted to larger Tilghman's plants.

Abrasive suited to the work is fed by gravity from a storage hopper into the wheel, and is then thrown by centrifugal force on to the work to be cleaned. The wheel is equipped with a device for controlling the direction of the abrasive. After use, the abrasive is returned by a scroll, fitted between the outer casing and the barrel, and is deposited into the boot of a bucket elevator, which returns the abrasive to the hopper ready for re-use.

The barrel rotates on four electrically-driven trunnion rollers at a speed of 2 r.p.m. An interlock is provided to cut-out the wheel motor when the barrel motor is stopped. The Rotary-blast will handle a load of 6 cwt and has a work capacity of 6 cu ft. Despite its modest dimensions, the Rotary-blast has a remarkable performance as regards output and cost economy. It is capable of cleaning a wide variety of components and is specially suitable for cleaning fragile pieces that are readily susceptible to damage.

Ease of maintenance has received the most careful consideration and the number of wearing parts has been kept to a minimum. In addition, the simple yet effective design features have an important bearing on the economical consumption of abrasive. The unit is supplied complete with exhaust fan and effective dust arrester, similar in design to the standard range of Tilghman's dust collector units.

NOISE REDUCTION

IN a paper entitled "The Sound-Proofing of Automobiles," in the *S.I.A. Journal*, January, 1954, the author, M. Cabarat, gives an explanation of acoustical terms and analyses sources of automobile noise. These noises are mainly due to vibration, and should be eliminated at their sources; they include engine noise from intake, exhaust, and timing gear; transmission noise from the gearbox, differential and driving wheels; tyre and body noise. The bodywork serves as a resonator for sounds generated by the engine, transmission and tyres.

Exhaust noise is a complex sound, the frequency of which corresponds to that of exhaust valve opening. Baffle type silencers reduce gas wave velocity, while absorption silencers affect its pressure; both reduce high-frequency noise fairly well. Absorption-expansion and absorption-reflection silencers are also reviewed. Principles similar to those of exhaust silencing are employed for intake silencing. Noise from the timing gear may be reduced by the use of special materials and helical gear teeth. Use of helical gears also reduces gearbox noise, while attention to the finish of differential gear teeth, or thermal treatment to augment their sound damping capacity is desirable. Tyre noise arises from contact of the tread blocks with the ground, and the resultant vibrations cause the body panels to resonate.

Reduction of body noise due to vibrations, set up by the engine, exhaust system, shock absorbers and springs, and transmitted through the chassis, may be effected by use of anti-vibration mountings for these components. To reduce aerodynamic noise, abrupt changes in section at windscreens and windows should be avoided, and other openings, such as sliding roofs, should be placed where least turbulence may be set up. Mountings for components subject to vibration should not be situated in the middle of flat panels. To prevent simultaneous vibration of all panels at the same frequency, the division of large panels into elements by ribbing is necessary. Ribs should be straight, perpendicular to the supported edges of the panel, and should not intersect.

The insulating properties of panels depend on the ability of the material to dissipate vibratory energy in the form of heat. Sheet steel is good in this respect. Heating, ventilating, or electrical ducts make complete insulation difficult. Sound-absorbent materials mainly attenuate reverberations, and may reduce noise level by 5-6 db, but their efficiency at low frequencies is poor. The author states that rubber or bitumen base sound-deadening materials limit the vibratory movement of a resonating panel and reduce its amplitude considerably. *M.I.R.A. Abstract No.* 6761.

ELECTRIC SPARK MACHINING

The Sparcatron Process for Working Hard Materials

BECAUSE of its great potentialities, the Sparcatron process of machining by means of an electric spark must be of great interest to production engineers. It provides a means of working hard metals that cannot be machined by edge tools, or even be readily ground by abrasive wheels. In fact, there is no metal in existence, no matter how hard or tough, that cannot be machined by the Sparcatron method. The equipment for the process has been developed and is manufactured by Impregnated Diamond Products Limited under licence from Sparcatron Limited. Burton Griffiths & Co. Ltd., Mackadown Lane, Marston Green, Birmingham have been appointed sole selling agents for Great Britain and Northern Ireland.

Basically, the process employs the tremendous energy contained in an electric spark to blast away particles of metal as it strikes the workpiece. It is the method of applying and controlling the sparks in such a manner that accurate machining operations can be performed which makes the Sparcatron process a practical and economical proposition. At present, development has been concentrated on the production of holes and forms where the component can be held stationary. In this field alone, the process is already being used throughout the whole range of the tool-making industry. It has been successfully applied to making and opening wire and extrusion dies, blanking and forging dies, the sinking of stamping die forms, producing form tool shapes, and on special work involving complicated shapes in hard and fragile materials. It must be stressed, however, that the process can be used for all standard machining operations, such as cutting-off, sawing, turning, milling, hobbing, broaching, drilling and trepanning.

Obviously, the most important application of spark-cutting is in dealing with cemented carbides and highly alloyed, heat - resisting materials. Another important advantage of the process is that it can be used for machining steel parts in the hardened condition. It is a feature of the process that such parts can be spark-machined without loss of hardness in the material, since the cutting action of the spark is instantaneous and the cooling effect of the fluid that is injected between the electrodes and the workpiece is so strong that loss of hardness by heating is precluded.

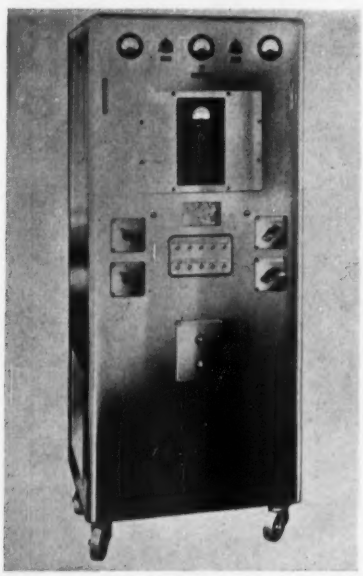


Fig. 1. The Sparcatron converter unit and control cabinet

Basic principles

An electric spark is caused by the sudden electrical breakdown of a dielectric (or insulator, such as air) separating two electrodes. Therefore, if the workpiece is made to be one electrode, and another electrode of opposite polarity is brought to within the breakdown point of the dielectric separating them, the ensuing spark will instantly remove a minute particle of the workpiece. It is evident that for a spark to occur at all, a gap must always exist between the two electrodes.

Under given conditions, two factors determine whether a spark discharge occurs or not; namely, the size of the gap and the potential difference, or voltage, existing across the electrodes. Another basic elementary factor upon which the Sparcatron process relies is that an electric spark will always "jump" across the shortest point of the gap between the electrodes. Because of this, whatever the cross section or contour of the electrode, the removal of metal from the workpiece will exactly follow the pattern of the electrode.

Although the original spark discharge may take place from any point on the face of the electrode, as soon as sparking occurs material is removed at one point on the workpiece and the gap there is then larger than elsewhere. Sparking must, therefore, take place from another point on the electrode, then another, and so on until the complete surface of the electrode has been covered. It is obvious that at this stage a minute quantity of material following the exact shape and topography of the electrode will have been removed from the workpiece. Theoretically, sparking will cease because the overall gap between the electrode and the workpiece will be greater than that required for a spark discharge under the prevailing conditions. However, if a practical means of maintaining the gap at the optimum value can be applied, and the generation of spark discharges made repetitive and continuous, metal removal will continue indefinitely until the hole is fully pierced or the desired sunken contour is reached.

Practical requirements

The fact that a gap, however small, must exist between the electrode and the workpiece, means that for a given size the electrode must be smaller on all dimensions by an amount equal to the spark gap. Also, for a given voltage difference between the electrode and the workpiece and for a given dielectric separating them, the spark gap must be maintained at the correct "breakdown" distance. Provision must also be made for generating properly spaced spark discharges in a continuous sequence.

Three practical factors must be considered. First, the length and intensity of the individual spark impulses and the length of the intervals between them must be carefully controlled to prevent the formation of a continuous arc. Secondly, provision must be made for controlling the intensity of the sparking effect over a wide range, because the surface quality obtained with this process (as with others) depends upon the cutting rate; the higher the surface quality required the slower the cutting rate must be.

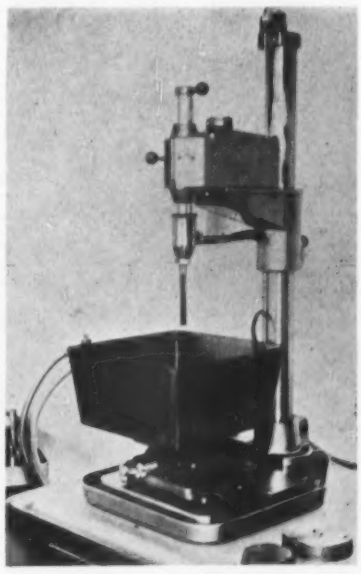


Fig. 2. Electrode holder, drill stand, work table and dielectric bath

Thirdly, the point of spark application must be kept under perfect control, because the accuracy of the resulting cavity is related to the spark gap. For example, the resulting diameter of a plain circular hole will be that of the electrode plus the radial clearance that constitutes the spark gap. As the accuracy of machining is so closely related to the spark gap and is therefore bound up with the electrical characteristics of the system, it follows that the spark gap should be kept as small as is compatible with operational requirements, including that of cutting speed.

The Sparcatron method

A simple relaxation circuit is the basic circuit used in Sparcatron equipment for generating the required train of spark discharges The energy for the spark discharge is derived from a condenser, which is charged from a D.C. source produced by circuits within the Sparcatron cabinet shown in Fig. 1. At the start of the cycle. the voltage across the condenser and the operating gap is initially zero, but it rises exponentially until it balances the supply voltage. However, the spark gap is such that the breakdown voltage of the dielectric medium is reached just before the condenser is fully charged. A spark occurs and the condenser is rapidly discharged across the gap. Therefore, the charge is damped and an arc cannot develop across the gap. The succeeding spark will occur only when the condenser is again recharged to the breakdown value. In effect, the condenser acts as an electrical valve, storing energy from the supply until the amount required to breach the spark gap is reached. This cycle is repeated automatically and sparks in rapid sequence.

Fig. 4. Arrangement for piercing a simple extrusion shape in hardened steel

Fig. 3. Set-up for spark-cutting a thread chaser from a solid piece of tungsten carbide

The amount of energy released in the spark gap when the condenser discharges is directly proportional to the capacity of the condenser and to the square of the applied voltage. There are, therefore, two means of controlling the intensity of the sparks and hence the rate of metal removal; by varying the voltage and by varying the capacity of the condenser. Sparcatron equipment provides two outputs, 110 volts × 10 amps and 25 volts × 100 amps, while a bank of separately switched condensers gives a wide range of combinations from 1 mfd to 612 mfd.

Control of the electrode to ensure that the correct spark gap is consistently maintained is carried out by an extremely sensitive servo mechanism. An idea of the sensitivity may be derived from the fact that the gap required when operating on 110 volts is in the order of 0·0005-0·001 in. To maintain such fine control, the servo mechanism is coupled electronically to the spark discharge circuit and functions automatically in relation to the discharge current. A signal voltage, proportional to the discharge current, is fed from a resistor in series with the electrode and is "compared" with a reference voltage. The reference voltage is so adjusted that with the electrode correctly positioned, the two voltages are equal and the effective signal voltage is zero. If the electrode deviates from this position, a positive or negative "error" signal is fed into an amplifier, the output from which energizes the D.C. servo motor. The motor is of the split field type and its direction of rotation depends upon the particular side of the field that is energized, which in turn depends upon the polarity of the error signal. As the motor shaft is coupled to a reduction gear that drives a pinion engaged with a rack on the electrode holder, the electrode is raised or lowered automatically in relation to the discharge current and the precise gap between the electrode and the workpiece is maintained. Certain refinements are incorporated to prevent the electrode "hunting" about the desired position.

In theory, spark gap machining may be carried out with the electrode and workpiece separated only by air, but in practice a liquid dielectric is used, principally for its cooling effect but also for certain electrical properties. Generally, the liquid employed is ordinary paraffin, although specially prepared liquids may be used. Despite its inflammability, the paraffin will not ignite, since the cutting point is fully immersed and the amount of oxygen present is insufficient to support combustion. The equipment includes a circulating system whereby the dielectric paraffin is pumped from a settling tank in the cabinet base of the machine into a dielectric bath surrounding the work table and back again into the tank. This method not only improves the cooling effect but also removes the fine sludge produced during the process. This sludge settles at the bottom of the tank, whence it can be removed periodically. Fig. 2 shows a Sparcatron servo-electrode holder, drill press stand, work table and dielectric bath.

The electrode may be fashioned from almost any soft material that forms a good electrical conductor; brass is generally recommended. Since brass is easily worked, almost any form of electrode, from a simple piercing tool to an intricate die, can be quickly produced. Furthermore, in many cases it is necessary to machine only the first electrode, since if the product is an

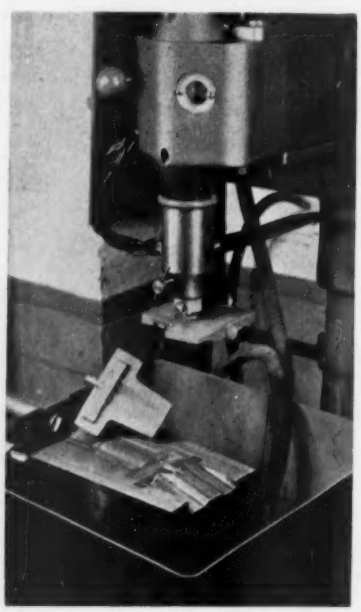


Fig. 5. Set-up for producing a forging die in hardened steel

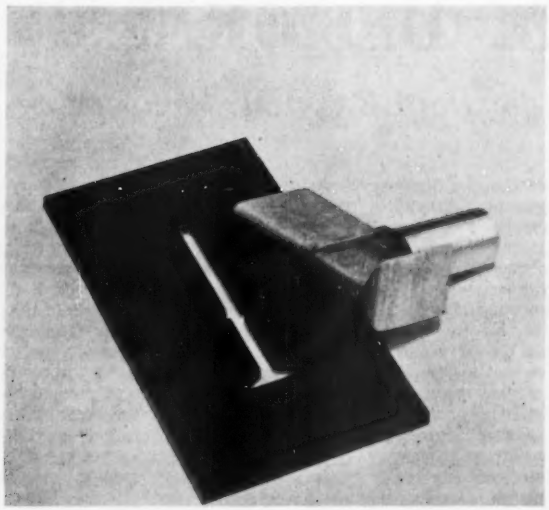


Fig. 6. Blanking die and the electrode used to produce it

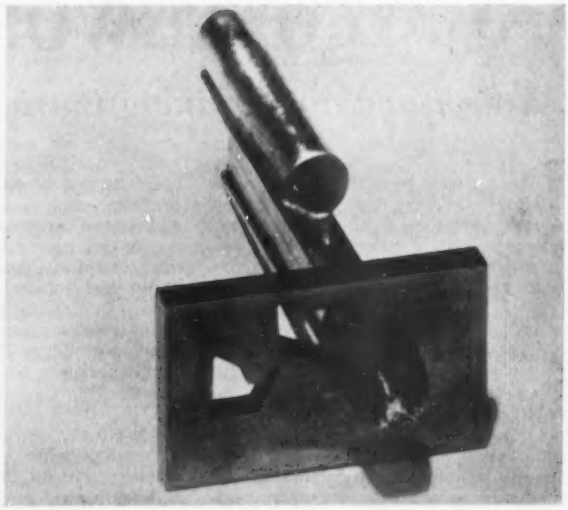


Fig. 7. Extrusion die in tungsten carbide and its electrode

extrusion or forging die, it is a simple matter to arrange for a number of pieces in brass to be produced from the original die to serve as future electrodes. In such cases a simple method of reducing the workpieces on all dimensions in order to provide the spark gap clearance is to dip them in a suitable reducing acid, thereby easily and cheaply converting them into electrodes.

Applications

Undoubtedly there are enormous potentialities for the Sparcatron process. For example, it opens up completely new fields for the use of cemented carbides. Hitherto, the use of these materials has in the main been confined to tool tips, press tools for long runs and wear resistant parts of simple shape, principally because of the difficulties encountered in machining. The Sparcatron process eliminates the machining problems. A set-up for producing a thread chaser from a solid block of tungsten carbide is shown in Fig. 3. It should be noted that for the sake of clarity the work clamps have been removed. The process is also particularly suitable as an alternative to grinding carbide tool tips, since it has the advantage of complete freedom from friction heat with the consequence that surface stresses and microcracks are completely eliminated.

This process also opens up possibilities in machining hardened workpieces. Through its use it is possible to completely finish-harden work, which can then be machined to size by spark-cutting without any fear of distortion. Applications on hardened steel are shown in Figs. 4 and 5. Fig. 4 shows the equipment arranged for piercing a simple extrusion shape in hardened die steel, while Fig. 5 shows the set-up for sinking one half of a forging die. A spare brass electrode is shown resting on the face of the die.

Other electrodes and workpieces are shown in Figs. 6 and 7. Fig. 6 shows a very high precision blanking die made in hardened 15 per cent tungsten steel. The length of cut is $3\frac{1}{4}$ in and the time taken was 11 hrs 33 mins. An extrusion die in tungsten carbide is shown in Fig. 7. This die is $\frac{3}{4}$ in thick and was produced in 2 hrs 55 mins. The power used was 100 watts and the capacity 6 mfd.

In many cases the standard electrode head and standard electrical unit may be used for converting existing machine tools to spark-cutting operations. In contrast with conventional cutting methods, the Sparcatron process has the advantage that the cutting electrode does not exert any mechanical tool thrust on the workpiece; therefore, heavy machine frames, tool slides and holders are not required.

The surface produced by spark-cutting has a non-directional finish and has special oil retention properties. Furthermore, the change-over from rough to finish machining merely involves the switching out of condensers and can be carried out instantaneously without any interruption of the cut. In this respect it is interesting to note that suitably stepped or tapered electrodes enable, for example, holes to be roughly pierced at a fast rate (using a high condenser value) from the first part of the electrode and then finish machined by the remainder of the electrode by simply switching to a low condenser value.

The power supply section of the main cabinet is sufficient to allow up to four spark-cutting heads to be operated. Separate auxiliary control units, comprising servo-control amplifier and condenser bank for each additional head, are being manufactured by Impregnated Diamond Products Limited. These compact units allow other heads to be added at much less cost than the original unit, up to a total of four. Burton, Griffiths and Co. Ltd., in collaboration with Impregnated Diamond Products Limited, will be pleased to give advice and assistance to those wishing to investigate the application of spark-cutting to their own specific machining problems.

CABMA REGISTER 1954-55

This new, improved edition of the CABMA Register of British Products and Canadian Distributors will be the Canadian buyer's standard reference work. A careful study of user reactions to the first edition has led to certain improvements in the 1954-55 edition, both in the arrangement of the various sections and in the quality of paper used.

Once again the Buyers' Guide provides an alphabetical list of British products available on the Canadian market. There are more than 3,750 headings and 1,300 cross references. This section is now followed by the French-English glossary, which gives the French equivalents of all the British product headings in the Buyers' Guide.

An alphabetical directory of British manufacturers and distributors gives details of the 4,500 British firms classified according to products. The details include the distribution arrangements in Canada, involving more than 2,750 Canadian companies. Many of these Canadian firms give further details in the Canadian Distributors' Announcements. Finally, there are Proprietary Names and Trade Marks sections. For easy reference, the six sections of the register are divided by index cards, with reinforced tabs and containing instructions in English and French.

This Register of 780 pages, bound in full cloth is published at 42s. 0d. (by post 44s. 0d.) jointly by Kelly's Directories Ltd. and Iliffe and Sons Ltd. for the Canadian Association of British Manufacturers and Agencies.

COACHWORK DESIGN

A Review of the Winning Entries for the I.B.C.A.M. Competition

IT has always been found that in the annual Drawing Competition organized by the Institute of British Carriage and Automobile Manufacturers, some entries are outstandingly good and well repay the closest attention of those interested in car design and draughtsmanship. Nevertheless, it is equally true that many entries are poorly executed and lack much of the essential information which the organizers require.

Over a period of years it has been found that some competitors start by submitting indifferent entries, but gradually succeed in producing work that warrants the highest praise; in this lies the great value of a competition such as that reviewed in these notes. The parent body responsible is supported by the Worshipful Company of Coach Makers and the Society of Motor Manufacturers and Traders, and the prize money allocated to the several sections is sufficiently attractive to appeal to a wide section of the trade. The organizers are at some pains to encourage competitors by specifying the types of vehicles, which range from the specialized custom-built body to the commercial passenger-carrying vehicle and the goods van. In addition, certain of the more important sections have a supplementary award that is most attractive, for it makes possible a visit to a Continental Motor Show virtually free of cost to the successful entrant. It is unfortunate that these competitions do not receive more support, but fortunately the organizers persist in their commendable aims, and it is sincerely to be hoped that this annual event will continue for many years.

In making the awards, the judges consider design, execution, practicability and originality. Each of these points can be given up to 25 per cent of the total maximum marks. It would be helpful to the competitor and of some value to the interested layman if an indication was given of the value which the judges place on each of the four points.

Competition No. 1

This called for a coloured drawing and also an outline drawing of a long-distance touring saloon of advanced styling, having comfortable seating for a minimum of four persons in addition to the driver. Special emphasis was placed on amenities for Continental touring, and ample accommodation for luggage was to be provided within the body. The coloured drawing, a three-quarter front view, was required to a scale of $\frac{3}{4}$ in to the foot. A scale of $1\frac{1}{2}$ in to the foot was to be used for the outline drawing, which had to provide essential information for checking practicability, particularly in regard to the arrangement of seating, luggage accommodation and convenience for touring. No chassis was specified, but the competitors were required to adhere to a wheelbase of 10 ft 11 in, all other chassis details being left to the discretion of the entrant.

The first prize of £50 and the Company's medal was awarded to R. D. Haynes, whose two drawings were executed in excellent style. The coloured drawing in particular attracted attention, for the body styling is shown to excellent effect and the draughtsmanship is of the highest order.

As befits a car of this type, the style is distinctly Continental, the rounded front panels being carried through on the body sides by the deep front wings which pass towards the rear with a gently falling line slightly below the lower line of the windows. There is no distinct break between the front and rear wings, since the extension of the front wing line is upswept in the region of the quarter window to rise to the maximum height of the rear wing, whilst in plan the overall wing width contracts to a corresponding point before sweeping outwards to provide the maximum rear wing width. To the rear of the wheel centre the rear wing, which is essentially an integral part of the body panels, contracts to the conventional width, and with the gently swept top panels provides a well balanced and pleasing rear end.

This rear end, although having the appearance of a somewhat elongated luggage boot, is in actual fact the housing for the 200 b.h.p. gas turbine engine. It has therefore been necessary to arrange for the disposal of luggage in less conventional places and the methods used by this competitor are ingenious. For example, beneath each of the front seats there is a fitted suitcase which can be stowed quite conveniently when the front doors are open. A similar arrangement is followed beneath the rear seat and here the designer specifies that the fitted suitcases are to be of the latest hanging wardrobe type, a type which is quite feasible because of the space available. Further luggage space is provided between the rear seat squab and the engine compartment, and as a matter

A reproduction drawing by R. D. Haynes for the winning entry, in Competition No. 1

of interest, this luggage hold measures 28 in × 21 in × 62 in, a compartment quite capable of housing a considerable quantity of luggage. Access to this luggage hold is through doors in the side panels; these doors are arranged to hinge upwards and one carries a drinking water container between the inner and outer panels.

From a coachbuilder's viewpoint this body has much to commend it, but it is felt that a chassis technician might be somewhat concerned at the weight which is being carried by the rear axle unit and tyres, the latter in particular being very highly stressed.

Since there has been no necessity to consider the arrangement of cool air entry at the front, it has been possible to style the bonnet panels to suit the well swept lines of the roof and rear wings, and the general effect is most pleasing. There is, of course, a small entry for cold air in the front panels to supply air to a filtration unit, whence it is passed to the interior of the car. Ducts up the screen pillars convey air from scuttle inlets to a roof lining outlet. From this same inlet air is released to provide demisting for the side windows.

No separate heater is fitted, heat being obtained from the turbine heat exchangers. The fresh air for this purpose is extracted from the main air intakes and is passed through the core of the exchangers and fed to the passenger compartment by two blowers driven by shaft from propellers situated in the exhaust gas stream. The warm air is ducted up the centre of the car and is released in the region where it warms the passengers' feet, and further streams of warm air are conveyed along the sides of the car to be released at each side of the screen to act as de-icers.

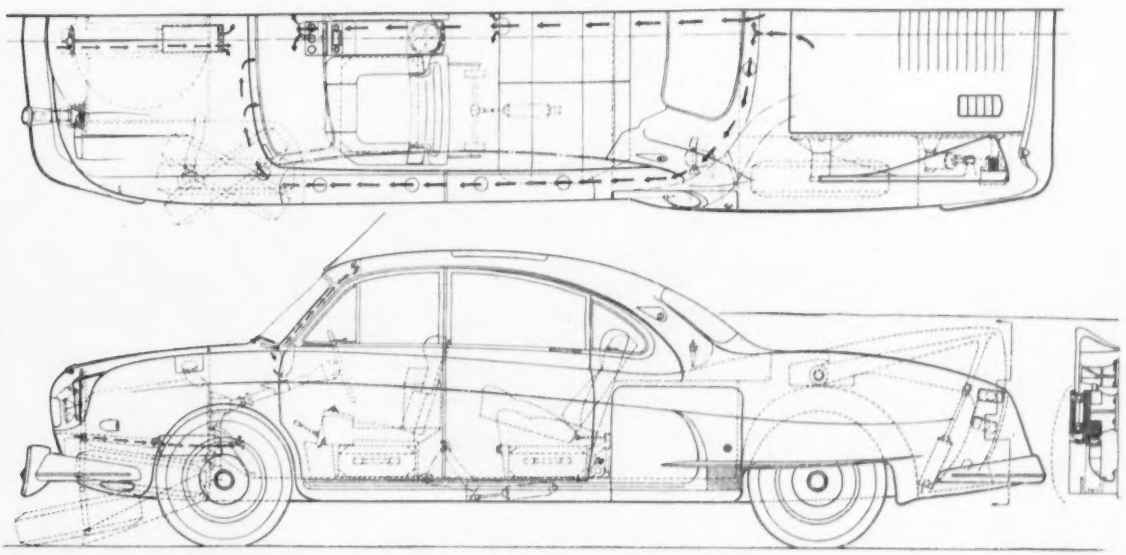
As regards the interior layout, there are two separate front seats which are separated by a compartment containing a box suitable for a map and other touring documents. The box lid can be folded over to make a convenient map table; a light is provided to facilitate this process. The radio set is also accommodated between the seats and it is interesting to note that this competitor favours a central location for the hand brake and gear control. Also dividing the front seats is a cocktail cabinet, which since it is arranged to rotate is accessible to both front and rear passengers. A device has been evolved that permits the cabinet to be disengaged from its turntable and removed from the car. This central cabinet also contains a clock and ash-tray, and it is mounted above a drawer housing food, crockery and linen. The back of each front seat is fitted with a folding table and the front doors contain large pockets for thermos flasks.

The rear seat squab is divided into three sections and the central portion is arranged to fold down, providing a table between the side seats and permitting a travelling bureau with a roll-front to be fastened in its place. This bureau when locked is secured to the rear squab board. When the car is carrying its full complement of passengers, the bureau is placed on the wide shelf behind the rear squab. Amongst the touring accessories that are specified are an electric kettle and teapot, and picnic stools are housed in the off-side luggage hold door.

The arrangement drawing is fully dimensioned to indicate the room available for passengers. To give a more restful seating position when desired, the rear seat is fitted with a device that tilts the squab and raises the cushion. The spare wheel is housed in a folding tray beneath the bonnet, the wheel being extracted beneath the massive bumper when required. Within the bonnet panels are housed the fuel tanks. They are of the plastic non-crushable type and can be filled from either side. Each of the two tanks has a capacity of 25 gallons. The 24-volt batteries are in the centre of the bonnet compartment, whilst the tool kit is arranged on the off side.

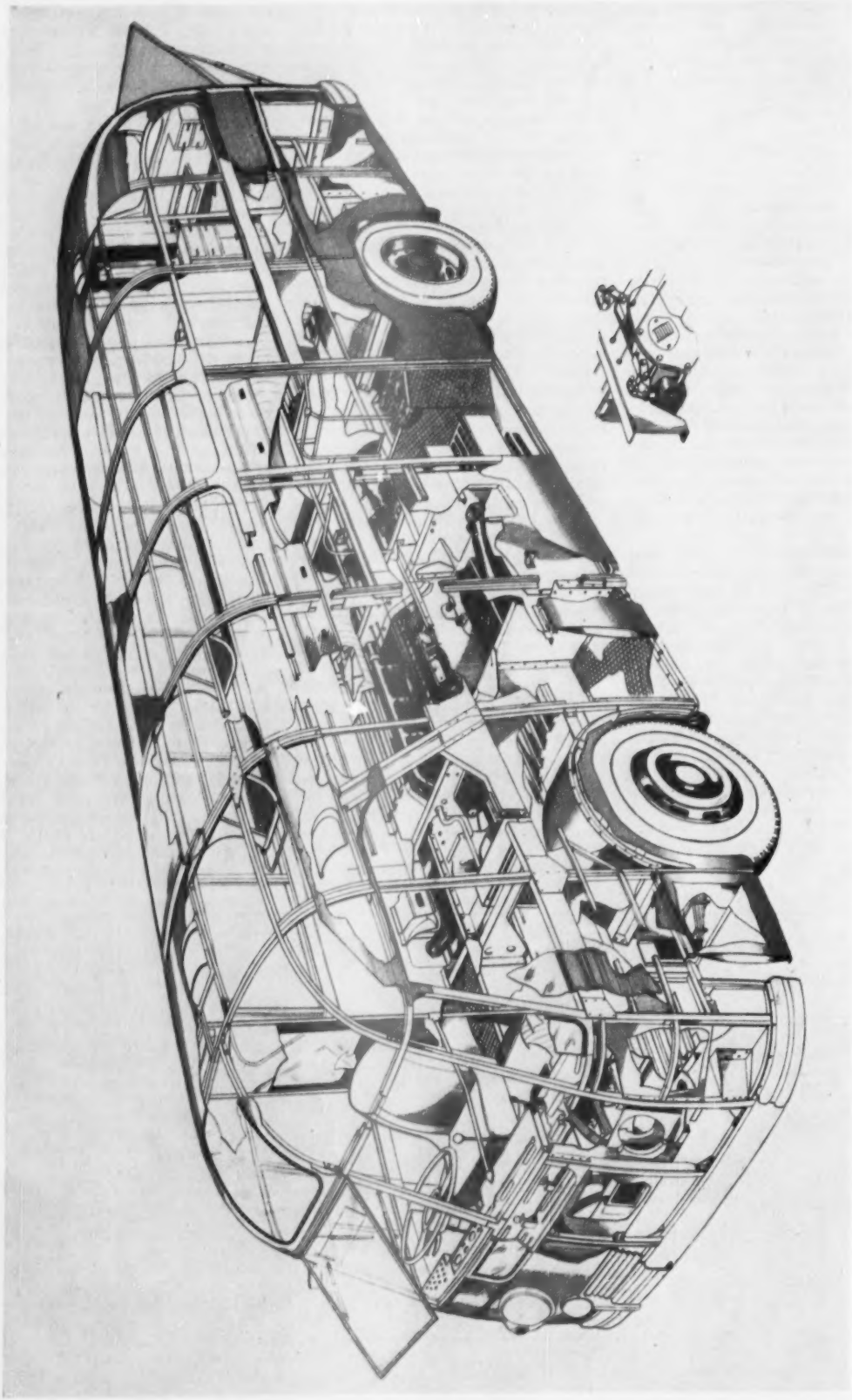
In this section there was no second prize award, but third prize was taken by D. C. Coward, who receives £15 and the Company's medal. This design quite definitely lacks the appeal of the drawing securing the primary award and the information shown is not nearly so complete. The design is characterized by the severe horizontal mouldings at the waist and also midway down the side panel. The sharp edge and somewhat angular treatment that is the predominating feature is not carried out in the contouring of the roof. The screen is very sharply inclined and, together with the side panels that extend well to the rear of the door hinge line, the entire screen unit is composed of a single piece of plastic material; thus its replacement would be a costly item.

This competitor also adopts for his design, a gas turbine power unit, which is mounted in the conventional manner and is combined with a front wheel drive transmission. The fuel tanks are mounted in the wings at the side of the scuttle and the spare wheel is carried in the boot on the off side. Owing to its length, for it extends well to the rear of the wheel, the luggage boot can accommodate a reasonable amount, but here again it is felt that there might easily be excessive weight on the rear axle and tyres. The seats are arranged in conventional style. At the front, there is one full-width seat divided into three sections. Since this vehicle has a steering arrangement giving a steering wheel in the centre, the centre portion has a more forward location than those at either side. Because of this



Layout of the design that gained first prize in Competition No. 1

L.



First prize in Competition No. 4 was awarded jointly to two competitors. One was D. G. Thaxter, who produced this design

arrangement it has been felt necessary to slightly stagger the front seating in order to improve elbow room for the driver; a commendable arrangement.

Certain special equipment for Continental touring is shown, but it is felt that much more is demanded in a car designed for Continental touring, and apart from the heavy outline, this drawing lacks clarity. The outline drawing is supported by a coloured drawing, which also reveals certain lack of experience in this class of work.

Competition No. 3

This was the Students' competition and was open to those studying at any technical school or to employees in British workshops, up to an age limit of 21. The competition called for an outline drawing of a two-door two-seater enclosed car of racing type, and details of the chassis were left to the discretion of the competitor although a wheelbase of 8 ft 3 in was specified. The drawing was required to a scale of 1½ in to the foot and could be in ink or pencil, and was to show the side elevation, half-front, half-rear and half plan views.

In preparing the design, competitors were required to adhere to certain conditions covered by the General Competition Rules issued by the R.A.C., and amongst these was the necessity for permanent mudguards, certain minimum dimensions for the screens, solid doors giving convenient access to the front seats, and a minimum area for the door windows. It was also required that the headroom above the seat cushions should be not less than 85 cms, and a rear view mirror of a certain size was to be included.

The first prize of £20 was taken by D. M. Morris with a pleasing design having well curved lines which blended with the frontal formation he adopted. With the low bonnet that would be the essential feature of any racing car, it was found possible to form integral front wings with practically no indentation between the three distinct panels, and in this assembly banks of louvres are provided in both upper and side panels.

A streamlined shape has been adopted for the body and the well rounded plastic screen is matched by a large curved plastic rear light. Since there are no rear seats, no consideration has been given to maintaining headroom at the rear and it has been possible to give a headroom of 36 in above the front seats and to slope the roof line with an agreeable curve to merge into the rounded contours of the rear wings.

The abridged specification which this drawing includes, features push-button locks, combined rear light and stop lights, an alligator bonnet and hinged seats for access to luggage. It is intended that the body be panelled in light alloy material.

The second prize of £15 was awarded to J. Packwood, whose design in general followed similar lines in that the well swept roof falls away in a gradual sweep to the tail end of the wings. The windscreen has single curvature and is inclined at a sharp angle. There is ample room for luggage behind the seat. The path taken by cold air entering the front grille is clearly shown, an internal deflector being arranged to pass the air through the radiator matrix and then through the oil cooler. The path taken by this cooling air embraces the brake drum region and the louvres in the sides of the wings are used to extract the air. By this means it is hoped that a build-up of pressure under the bonnet will be avoided.

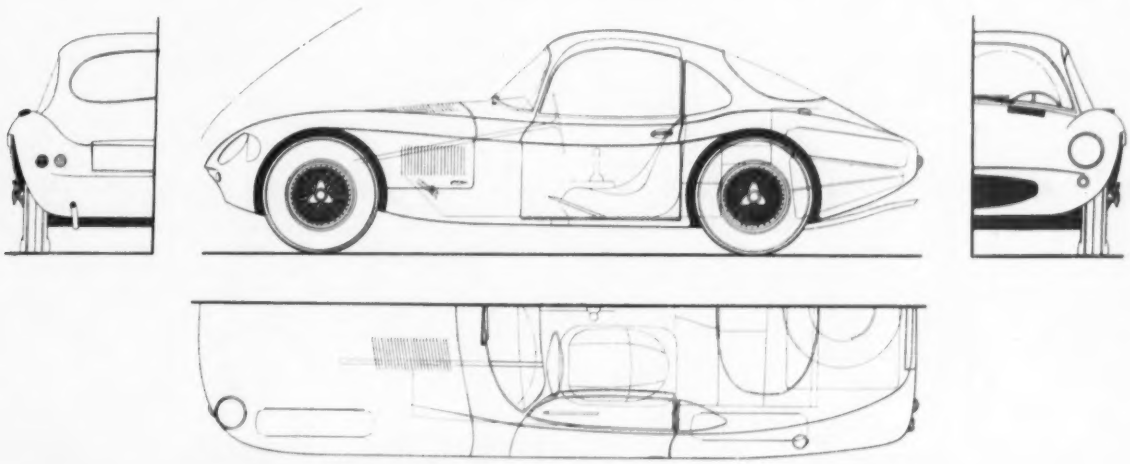
Considerable attention has been given to the question of air drag and to obtain a smooth exterior shape, the back light, quarter lights and head lamp glazing are embedded in rubber which makes the perspex panel form a continuous surface with the surrounding aluminium body panels. Somewhat unconventional fuel tanks are indicated. It is felt they would be extremely costly and difficult to make, those at the rear for example being curved to follow the shape of the wheel arches. Obviously such a tank would require an elaborate system of feed pipes in order to extract the last drop of fuel.

The third prize of £10 was awarded to P. Ludford for a somewhat similar design, a slight difference being that the rear extremities of the rear wings are very much deeper than those shown in the other successful drawings in this class. There is a large back light, curved in two planes, and the windscreen has similar contours. The front wing line is more distinct and merges into the body panels at the scuttle. The shape of the front door hinge line is unusual but the extension of the door on to the scuttle is obviously intended to provide easier access for the passengers and is to be commended on that account. This competitor also pays some attention to the extraction of air that enters through the cooling grille, and in this design the bonnet panels are arranged to open forwards and upwards.

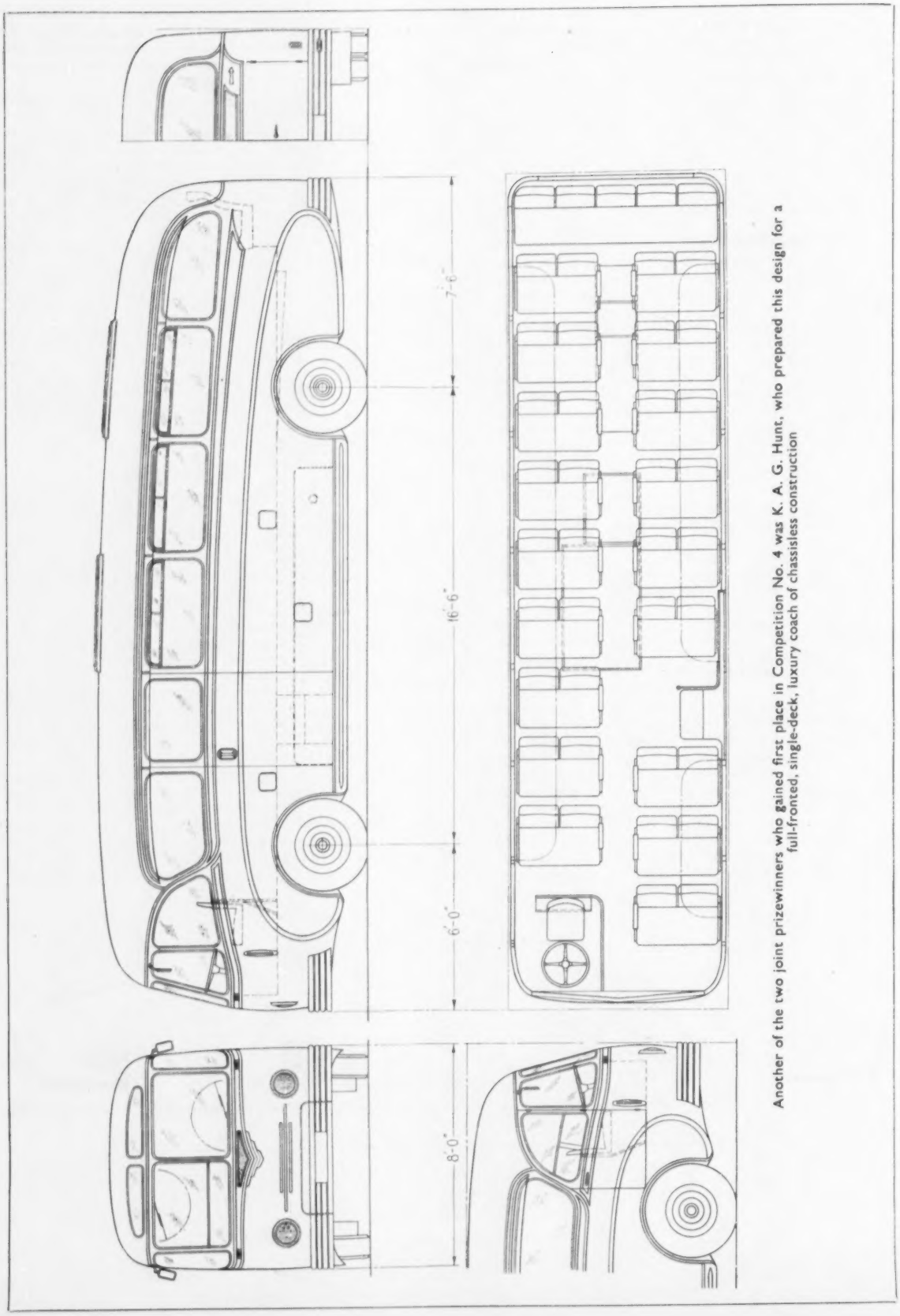
Competition No. 4

This competition was open to persons of British nationality, without age limit, and was for a general arrangement drawing, in ink, of a full-fronted single deck luxury coach of chassisless construction. The vehicle was to be suitable for general use by a British coachbuilder and its overall dimensions were to be 30 ft long by 8 ft wide. Naturally, to be of value to an operator in this country the vehicle had to comply with all current Ministry of Transport regulations appertaining to single deck vehicles. The drawing was required to a scale of ¾ in to the foot and was to show the side elevation, front, rear and plan views, and a half cross-section, giving details of construction employed. It was to be to a scale not exceeding 1½ in to the foot.

Particular attention was to be given to the method of installing the engine, transmission and all other units, and these methods were to be clearly shown.



The design, by D. M. Morris, which was awarded first prize in Competition No. 3



Another of the two joint prizewinners who gained first place in Competition No. 4 was K. A. G. Hunt, who prepared this design for a full-fronted, single-deck, luxury coach of chassisless construction

It was left to the discretion of the entrant to decide which type of unit to employ, bearing in mind prevailing manufacturing facilities. The number of passengers to be accommodated was not specified but it was pointed out in the conditions that maximum seating was to be achieved and there was to be adequate luggage accommodation for the number of passengers carried.

In making the awards in this section, the judges found difficulty in selecting one drawing as being superior in design and execution, and they decided that the first prize should be awarded jointly to two competitors. Accordingly, K. A. G. Hunt and D. G. Thaxter were each awarded £56 5s. and the Company's medal.

Part of the drawing by E. I. Owen who won third prize in Competition No. 4

The design submitted by K. A. G. Hunt followed most attractive lines, which are accentuated by curved mouldings at cant rail and waist. Both front and rear panels are agreeably swept to give at the front, a clean entry, which in terms of frontal overhang accounted for 6 ft of the total length. This design is particularly outstanding for its arrangement of the side windows flanking the screen. Immediately to the rear of the slightly V'd screen there is a curved glass leading on to the side of the body, while the side window, which follows, has a rearward edge that is inclined forwards. The main purpose of this is to suit the emergency door, which is arranged in the off side panel approximately in line with the driver, where it is thus available for both the driver and the passengers.

The entrance door is on the near side. Forward of the door there are three double seats. All the seats in this vehicle are of the forward facing type. Apart from the seat for five passengers across the rear of the body, all seats are arranged to take two passengers and the total seating capacity of the vehicle is 41. Sliding mechanism is adopted for the entrance door and the floor is recessed to provide two steps each 1 ft ½ in. The right-hand side of the step recess and the floor above it are provided with a decency screen and a guard rail.

It is interesting to note that the roof is equipped with two Weathershields hinged panels of the type that have been very well received by coach operators and builders, since, apart from the ventilation and extra lighting which they give, this arrangement can also provide useful means of exit in an emergency.

In the arrangement of the power unit and transmission and other details normally associated with a commercial vehicle chassis, the power unit is arranged on the off side approximately midway between the front and rear wheels. It is a Gardner horizontal oil engine having a maximum output of 94 b.h.p. at 1,700 r.p.m. A five-speed David Brown gearbox is used and the Borg and Beck clutch is complete with Lockheed hydraulic control. For the suspension, laminated semi-elliptic springs are used at front and rear, and on the front axle only Newton and Bennett hydraulic shock absorbers are fitted.

For the rear axle, this entrant has selected an Eaton two-speed unit, the speed selection being electrically operated. Braking is by triple servo vacuum mechanism of the Girling type assisted by Clayton Dewandre servo equipment. To complete the specification of the running units there is a Marles double cam and roller steering mechanism, a 12-volt electrical system and a Norman Isherwood contra-flow radiator. The fuel tank is mounted on the near side and has a capacity of 30 gallons.

In the construction of the body, light alloy plays a prominent part and extruded sections are used for the pillars, crib rails and cant rails. The interior stress panels are of 18 gauge sheet, whilst the additional stress in the region of the wheel arches is taken by 14 gauge sheet material. The exterior panels are in 19 gauge aluminium alloy sheet and 14 gauge material of the same specification is folded to provide wait rails and roof sticks. Window pans and shrouds are in pressed light alloy and in the assembly of the body framework, solid and Chobert type rivets in aluminium alloy are employed. Wherever they are required, high tensile steel bolts are used. It is interesting to note that for the floor panel laminated wood ⅜ in thick is used.

For convenient access to the several running units there are ample floor traps. Access to the engine is further simplified by a large detachable flap arranged in the off-side body panel and extending most of the distance between front and rear wheels. The rear luggage locker has a capacity of 91 cu ft. Access to this compartment is through double doors in the rear body panel. These doors are hinged at the sides and controlled by one central handle.

The pillars are of rectangular section and above the waist the width is built up from 1⅛ in to 4½ in by adding aluminium alloy window shrouds which in turn accommodate the Clatonrite glazing rubber. On the inside these window shrouds are trimmed in scratch-proof rexine, whilst the corresponding window pans on the exterior are finished in accordance with the exterior scheme. The folded waist rail forms a deep channel section having an outer depth of 3⅛ in; the inner flange, which takes the fixing of the stress panel, has a depth of 1½ in. The distance between the inner and outer panels is 1¼ in.

Apart from the general arrangement drawing and the cross section through the body giving pillar, bottom frame and rough constructional methods, this entrant includes a third sheet that clearly indicates the methods to be followed in setting out the underframe. This sheet also gives a very clear idea of the disposition of the running units and it is obvious that this successful entrant has given considerable thought to the question of vehicle maintenance. His drawing is clearly executed and leaves nothing to the imagination. This competitor estimates that the weight of his complete vehicle would be 5 tons 17 cwt 3 qr.

The design submitted by D. G. Thaxter which shared the first award with K. A. G. Hunt, was of a more elaborate nature in that it included a toilet compartment in the off-side rear corner and arrangements, also at the rear, for preparing light refreshments. For this reason the seating capacity is reduced to 30 passengers and it is felt that this design lacks the good appearance which characterizes the other design sharing this award. It is, however, a layout that well repays examination. Externally the body is characterized by the stepped-up waist line which rises at a sharp angle above the rear edge of the front wheel, thus the first two rows of seats have their backs well above the waist line and the range of vision for these passengers is excellent. Throughout the body the floor is level and all seats are arranged to face forwards, double seats being used with the exception of the last row which, to give access to the toilet and buffet have single seats at each side. The near-side folding seat has a folding flap to accommodate a courier or hostess.

The entrance door is of the sliding type and is immediately forward of the rear wheel, the floor being recessed to provide three steps; there are hand rails at either side. Curved glasses are fitted in the border panels of the roof above each of the main side windows, except those having the lower waist line, and these curved glasses are equipped with concertina blinds to prevent undesirable sun glare. In addition, Weathershields hinged roof vents with tinted perspex panels are used.

This drawing, in addition to the outline details called for, includes a coloured drawing finished in two shades of blue, the panels above the waist line being in light blue and those below the waist moulding in a darker blue. It is obvious that this competitor needs more experience in the preparation of this class of drawing, but his arrangement drawing is extremely well done.

In the construction of the body the vehicle is to be completely metal framed with mild steel pillars, underframe members and truss panels. In this way it is intended that a hollow beam shell of considerable torsional rigidity should be erected and timber packings are to provide fixings for panels, mouldings and interior finishes. The mechanical components are the same as those used in the production of current Leyland "Royal Tiger" chassis, with modifications as required for a chassisless vehicle. The Leyland engine is located beneath the floor and access is through a trap in the gangway. There is also a gearbox trap adjacent to the entrance steps. Heater ducts extend along both sides of the body at floor level and the roof panels are equipped with "Roevac" extractor ventilators.

The third prize of £15 and the Company's medal was awarded to E. I. Owen, whose design is for a 41-seater vehicle. This is a most attractive design, somewhat severe in general form but pleasing on that account, since the same theme is followed throughout. There is a pleasing absence of falling lines and the long rectangular side windows have well radiused corners, as also have the curved lights which are used in the rounded roof quarters.

This entrant adopts the entrance door in the near side panel of the cab and to gain access to the central gangway the passengers pass into the vehicle towards the driver's seat and then turn towards the rear to pass between the symmetrically arranged double seats. All seats are of forward facing type except those across the rear end of the body, and the layout adopted is quite a conventional one. The emergency exit is in the off-side panel adjacent to the driver's seat. On the near side between front and rear axles the side panel incorporates a luggage locker with a capacity of 40 cu ft, whilst the spare wheel and tool kit, together with the battery compartment, is also arranged on this side. A second luggage compartment is in the rear portion of the body and has a capacity of 62 cu ft; thus the total amount of luggage which can be disposed of is considerable.

In this body also aluminium alloy is relied upon for the construction of the body shell and the H-section pillars maintain this section from the skirting rail to the cant rail, above which point they are milled to T section. Extruded aluminium alloy is also employed for the underframe and other main members of the body, whilst for the inner and outer panel aluminium alloy sheet is used. In this case also both solid and Chobert type rivets are adopted for assembling the body members. All this information is very clearly indicated in a supplementary drawing that shows a cross section view of the body.

In a perspective sketch which is sectioned to show some of the constructional features, this competitor has very clearly indicated the method he favours for strengthening the floor, which embodies a recessed gangway. His methods are interesting, for support is obtained by adopting a form of girder construction that extends from a central unit running the length of the vehicle and braces both sides of that portion of the floor upon which the seats are mounted. The main strut, extending between the central member and the side framing of the body, carries diagonal members that have a bearing at their upper ends on a metal member running transversely; this is an interesting method of bottom frame arrangement. It has the advantage that adequate strength is available for the support of the floor and the body side framework without interfering with the space that is required for luggage and engine disposal. A further drawing indicates the layout of the major running components, showing the engine on the off-side, conveying the drive to the rear axle through a diagonally disposed shaft.

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Competition No. 5

A 10 cwt delivery van was the subject of this competition and a general arrangement drawing in ink was required. The body was to be designed for a chassis of forward control type and the payload capacity was not to exceed 150 cu ft. Special emphasis was to be given to the vehicle's suitability for large scale production, economy in tooling costs and practicability for shipment in a convenient form and assembly abroad. The drawing was required to a scale of 1½ in to the foot, and was to be in ink. Furthermore, sufficient information was required to indicate the various pressings used, the types of component and panel joints adopted, together with the usual full size sections through various parts of the body. Originality of styling and driver's comfort were given special consideration by the judges and convenient access to both the driving and payload compartments was emphasized. The selection of the chassis was left entirely to the entrant.

In this class the first prize was withheld but the second prize of £25 was taken by G. J. Clarke, who submitted a pleasing design of delivery van having a Brady self-aligning roller shutter in the rear panel Entrance to the driving compartment is through sliding doors on either side of the cab. The partition behind the driving seat extends the full width of the body and has a sliding near side half incorporated to enable the driver to gain access to the payload compartment without leaving the vehicle.

As far as possible this competitor uses pressings that embody components which in the past have often been separate units. For example, the rear wings are formed as part of the side panel pressings. Similarly, the advertisement panel above the waist line is pressed inwards and much has been done to simplify constructional procedure. The inclined screen has rounded corner panels which are followed by triangular windows, and it is noticed that the door is fitted with sliding glass. The floor is arranged at a convenient height of 1 ft 8 in above the ground and the laden overall height of the vehicle is 6 ft 6 in.

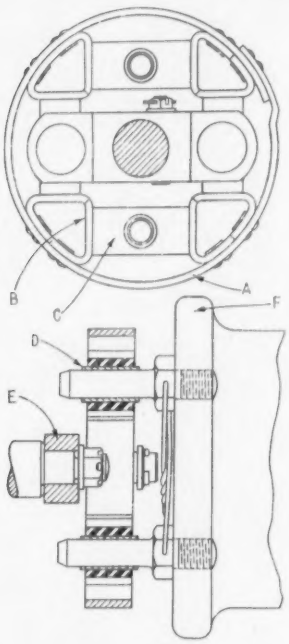
Economy of tooling is effected by the use of identical front and rear roof dome panels and the border panels are also identical. The screen is slightly inclined and is rounded at the corners to lead from the front surface to the side panels. The comfort of the driver is ensured by a Cox pedestal type seat which is adjustable vertically and also longitudinally. Two exterior driving mirrors are shown and the three-quarter front view which this competitor includes gives an excellent idea of the general appearance of the vehicle.

CURRENT PATENTS

A Review of Recent Automobile Specifications

Flexible coupling

LIKE the familiar Oldham coupling, this constant velocity device can accommodate limited axial displacement of driving and driven shafts but it is less susceptible to wear and noise when operated at high angular velocities. It consists



No. 705074

of an outer metal ring A, to the interior of which are attached four abutments B having flat faces at 90 deg. Between the faces of adjacent abutments are bonded rectangular blocks of rubber C each with a metal bushing D bonded at mid-position with its axis parallel to the shafts.

This assembly is engaged on two pins rigidly secured to member E attached to the driven shaft and is retained by washers and split pins. Alternate bushings are engaged on two pins screwed into the flange of the driving member F. *Patent No.* 705074. *S. Smith and Sons (England) Ltd.*

Upper deck for public service vehicle

CONSTRUCTED of light alloy, this deck includes main transverse girders A secured to the vertical pillars B of the side framing. Each girder comprises a flat plate with pairs of angles C riveted to its upper and lower margins. Joining the girders are two rows of longitudinal members D, so positioned transversely as to receive the inboard bearers for the seats. Further longitudinal members are constituted by angles E secured to the side frames.

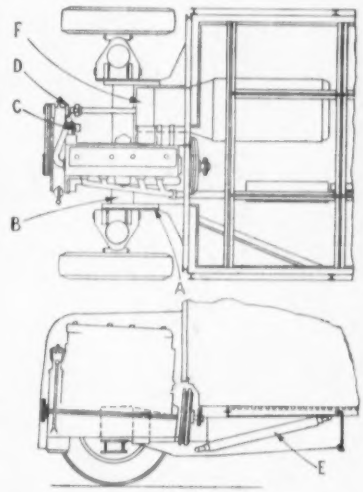
Additional transverse supports F, midway between girders A, each consist of a flanged vertical plate extending from a longitudinal vertical plate G, defining a channel along the side of the vehicle, to a member D. Midway between girders A and adjacent supports F are transverse members H of top-hat section following the contour of the lower edge of girder A and serving for the attachment of ceiling panels. The decking is built up of panels J each comprising a corrugated sheet sandwiched between two flat sheets, the corrugations being filled by strips of expanded plastics material for sound insulation. The panels are supported on the lateral flanges of girders A and longitudinals D and E and located by the vertical elements of those members.

To increase the strength of the floor in the central area forming the gangway, additional panels K having their corrugations running longitudinally are laid below the deck panels between the ends of members H and supported on the lower marginal angles of girders A. Fairings L are provided between the surface of the deck and the vehicle sides. *Patent No.* 704363. *British Transport Commission.*

Engine installation for double-decker

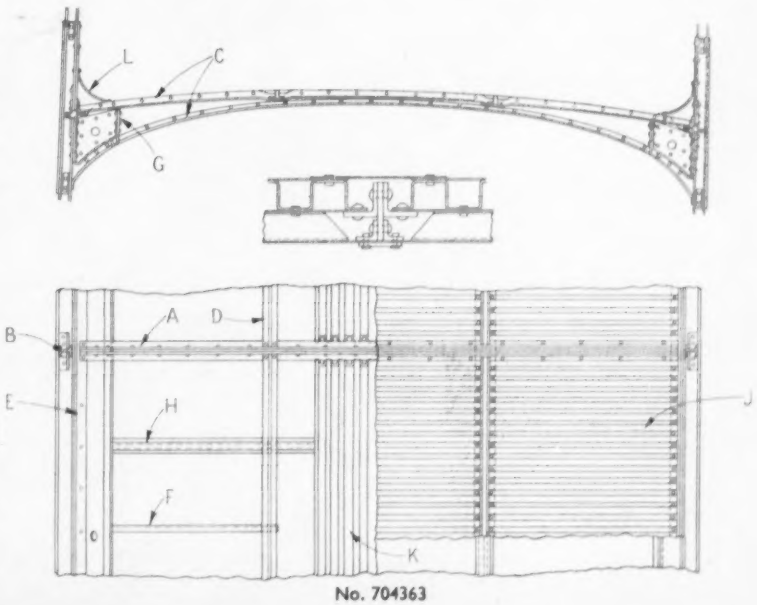
FOR public service vehicles the underfloor location of the engine is deemed undesirable since it entails some reduction of accessibility, complication of controls and structural interference with the low floor height necessary for double-deckers. This arrangement of the power unit and its auxiliaries aims to reduce the length of the space occupied by a forward-mounted vertical engine to a minimum.

In a chassisless vehicle two cantilever arms A extend from the front bulkhead to furnish attachment points for the independent wheel suspension units and bearers B on which the engine is resiliently mounted. On the front of the engine is a bracket carrying a coolant pump C and a spindle D driven by twin belts from the engine crankshaft. The rectangular radiator E is mounted on the offside below the floor of the passenger compartment and its upper side is enclosed in a cowling with a forward-extending neck carrying a fan housing F. A universal-jointed shaft transmits the drive from



No. 704562

spindle D to the fan. Air may be drawn into the rear of the cowling through ducting from an intake in an elevated position, to avoid contamination by dust or fumes, and delivered by the fan to ducts leading to warm-air discharge points in the passenger space. *Patent No.* 704562. *British Transport Commission.*

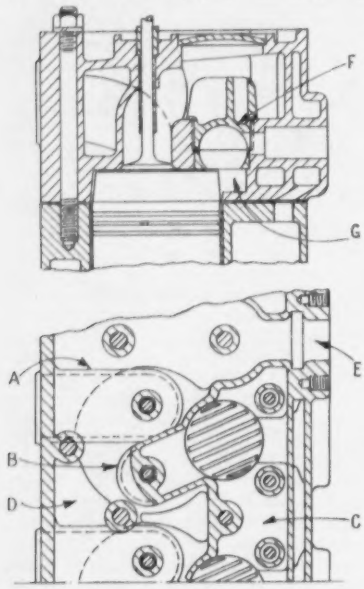


No. 704363

Compression-ignition engine

IT is suggested that the current practice is to design a compression-ignition engine with a higher compression ratio than strictly is necessary. The reason is to obtain as short a time lag as possible between injection and ignition of the fuel in order to avoid knocking or rough running. Accordingly, it is the aim of the invention to provide an engine that will operate efficiently on a compression ratio lower than is usual.

In the example shown, the inlet passages A communicate with apertured facings on the side of the cylinder head to which an induction manifold is



No. 704236

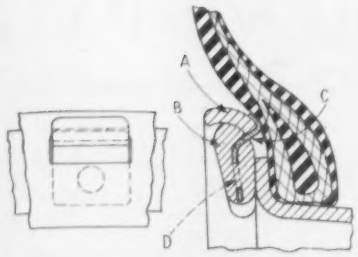
attached. The exhaust passages B, however, open into a space C segregated from the coolant space D by a wall and leave the head by a common outlet at the centre of the head on the side opposite to the inlet manifold and immediately below the coolant inlet E.

The combustion chamber is formed by a part-spherical cell in a hot body F communicating with a lateral cavity G. A large proportion of the heat of the exhaust gases can be transferred to the body F, by way of the vertical fins. A coolant jacket on the exhaust side of the head surrounds the boss in which the fuel injector is mounted and is in communication with inlet E and the cylinder jacket.

Despite a relatively low compression ratio—which may be substantially of the same order as that used in a spark-ignition engine—the preheating of the charge of air in the combustion cell ensures the ignition of the usual diesel fuel. The principle may be applied to either two-stroke or four-stroke engines, with either side or overhead valves. To enhance the heating of the combustion chamber the cylinder head may be of aluminium alloy and it may be covered with a heat-retaining jacket of asbestos or similar material. *Patent No.* 704236. *A. F. Sanders.*

Balancing road wheels

TO facilitate wheel balancing the flange A of the wheel rim is pierced at four points pitched at 90 deg around the circumference. A balance weight B is



No. 703044

shaped to fit snugly inside the flange and furnished with a cast-in, hook-shaped tongue C anchored by means of the hole D. The tongue is inserted through the opening in the flange and abuts the radially outer margin of the opening simultaneously as the weight engages the inner face of the flange. In this position the hooked end of the tongue is held firmly to the flange by the pressure of the inflated tyre, thus securing the assembly. The balance weight is so proportioned that it lies within the width of the flange and is not likely to be displaced should the wheel strike an obstruction. *Patent No.* 703044. *Daimler-Benz A-G.*

Multi-cell vehicle structure

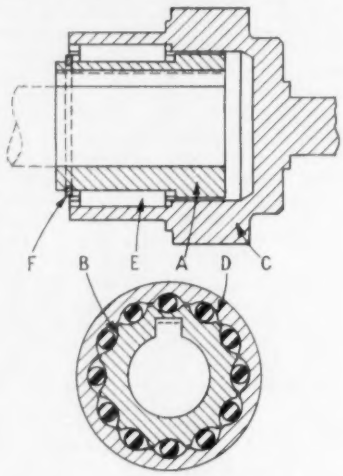
THIS invention relates particularly to the floor of the middle cell of a vehicle to which front and rear cells, preferably identical, are attached. The floor consists of a sheet-metal plate A with longitudinal and transverse edges bent upwardly and welded to a trough shape. From the plan it will be seen that the ends of the bowed transverse flanges B are bent outwardly, avoiding acute angles, to meet the projecting ends of the longitudinal flanges C. Angled sheet-metal parts D and E are welded into the trough to form a box-section ring frame.

End walls F welded to flanges B have an inturned marginal flange, the ends being welded to flanges C. To the outer face of each end wall and flange B is welded a boxed bearer G to which the respective end cell H is connected. Profiled strips J are either screwed or welded to flanges C and, with strips K on the end cells, form a protective band around the vehicle superstructure. *Patent No.* 703889. *Daimler-Benz A.G.* (*Germany*).

Torsionally resilient coupling

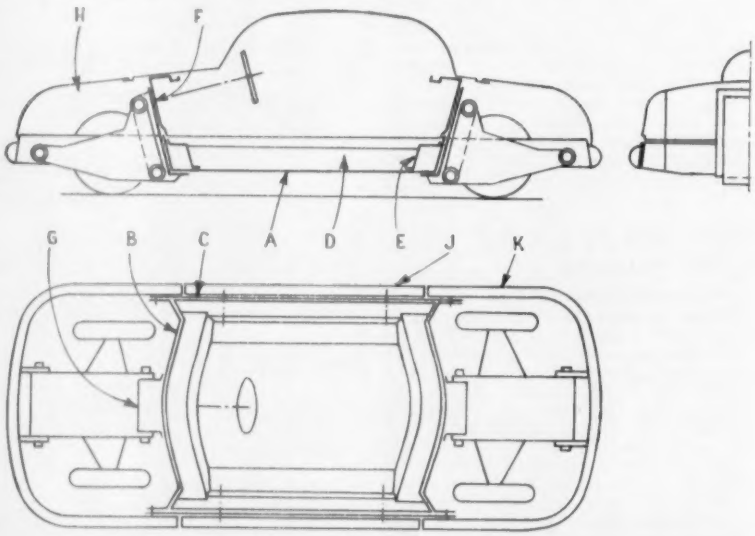
THIS coupling is intended to ensure a smooth take-up of drive at starting and to damp out torsional fluctuations during transmission of power. Sleeve A keyed to the driving shaft is furnished on its periphery with a regular series of arcuate recesses B. The driven member C, formed directly with or secured on the driven shaft, has an axial cavity provided with recesses D of similar shape and corresponding pitch to recesses B.

In each space formed by the co-operating recesses B and D is inserted a rod E of rubber or like material having a high elastic limit. They are housed with a



No. 704497

slight clearance to accommodate possible misalignment of driving and driven shafts. A spring ring F seated in a groove in sleeve A ensures their retention. *Patent No.* 704497. *Société des Fabrications Unicum* (*France*).



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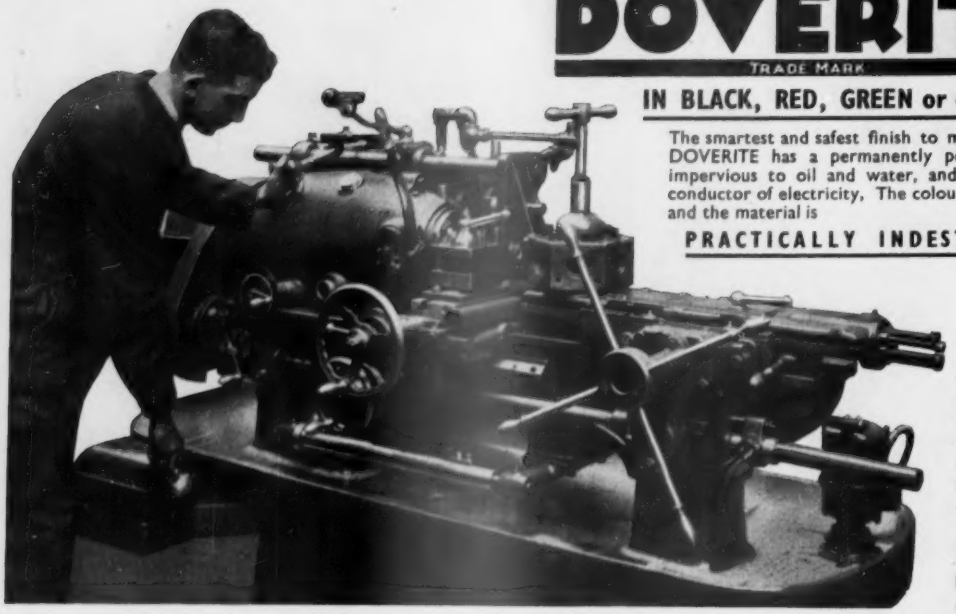

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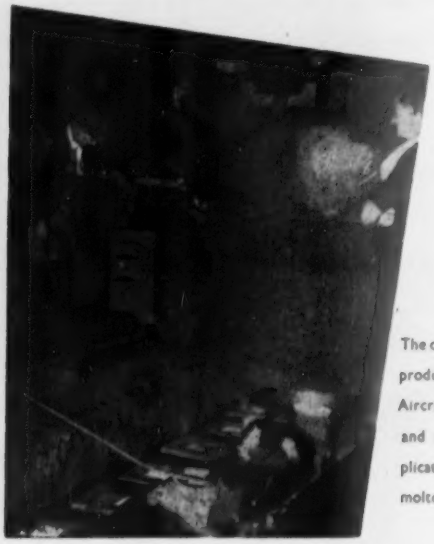

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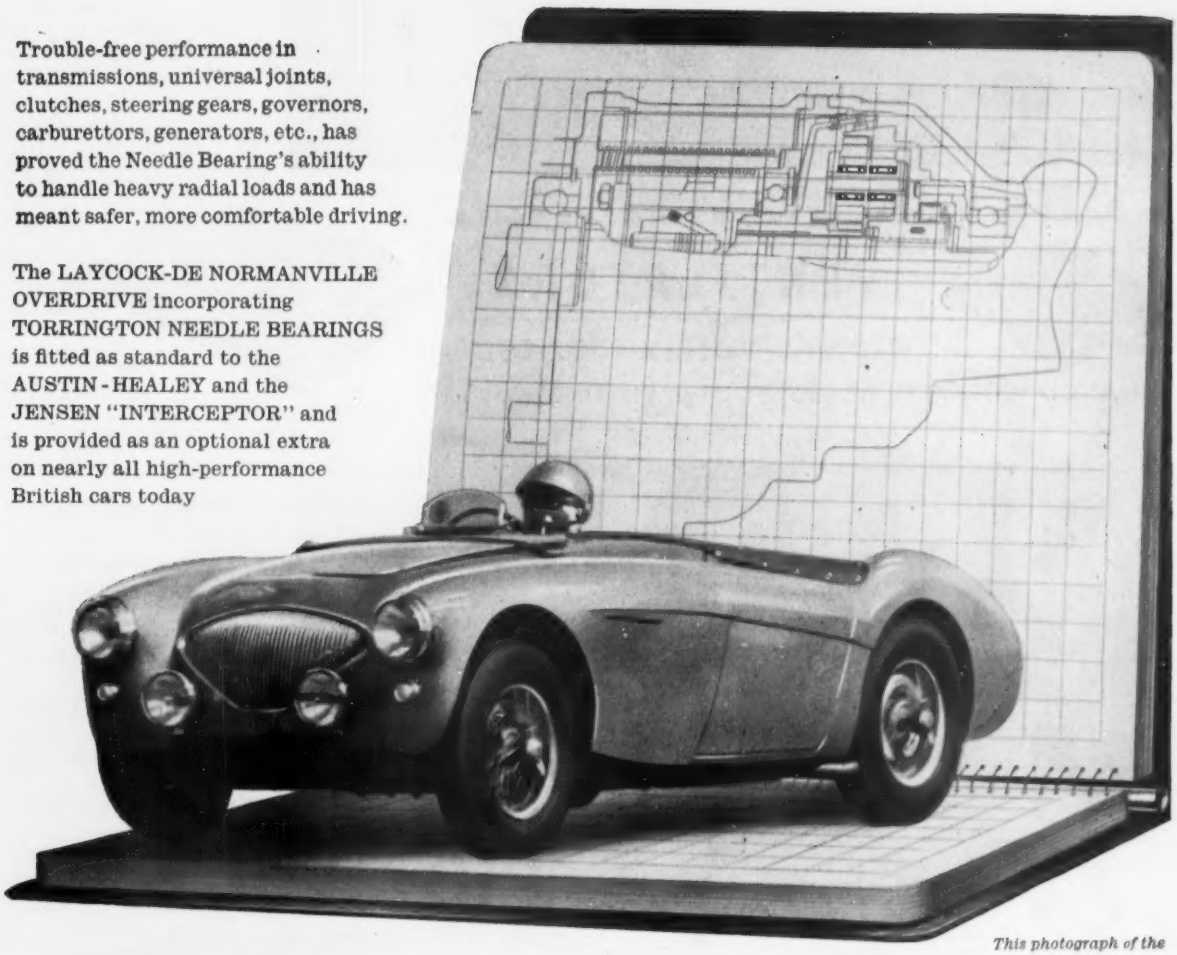

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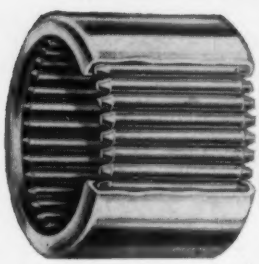

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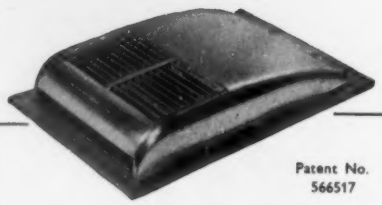

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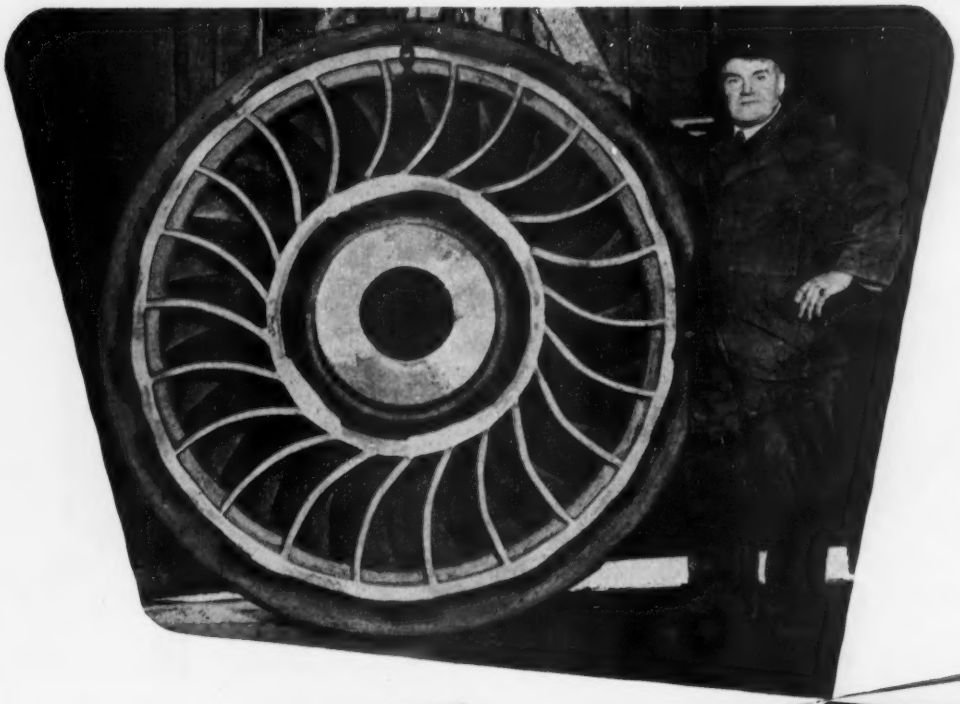

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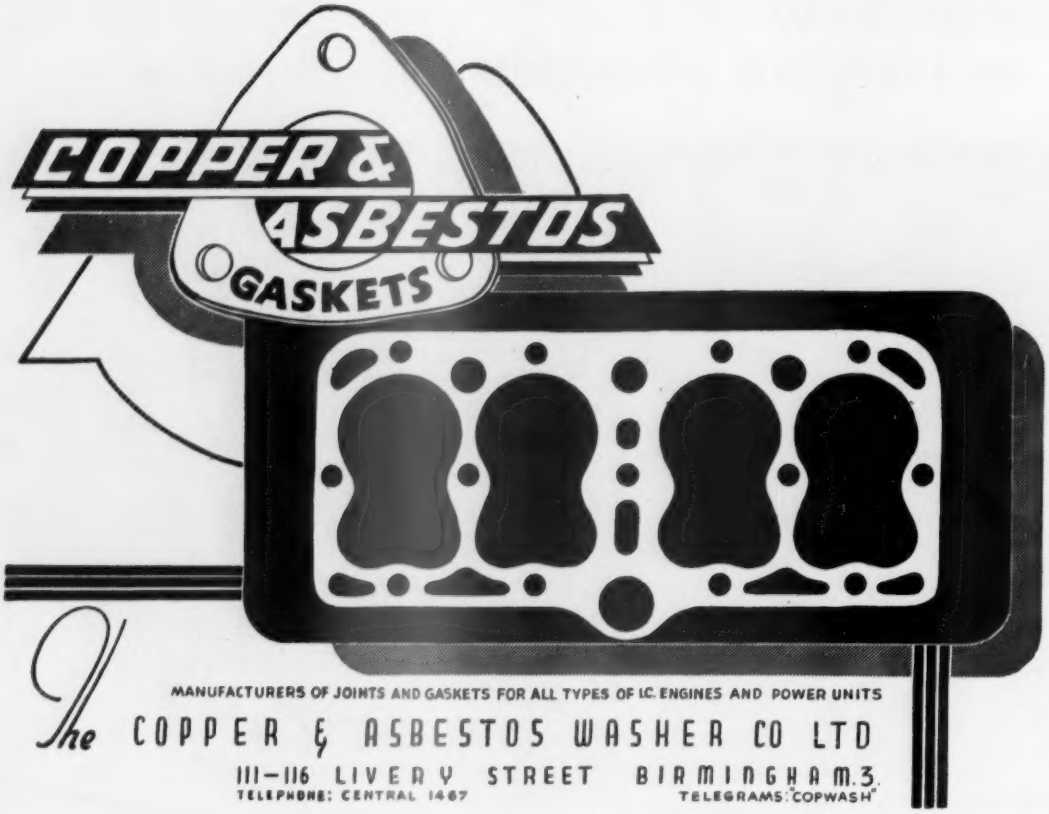

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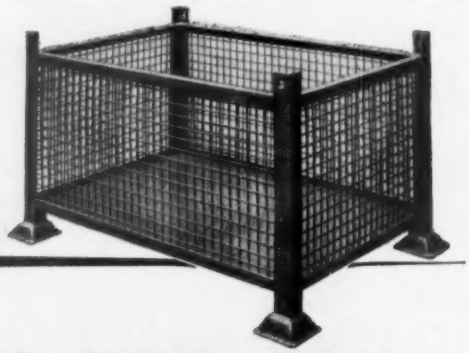

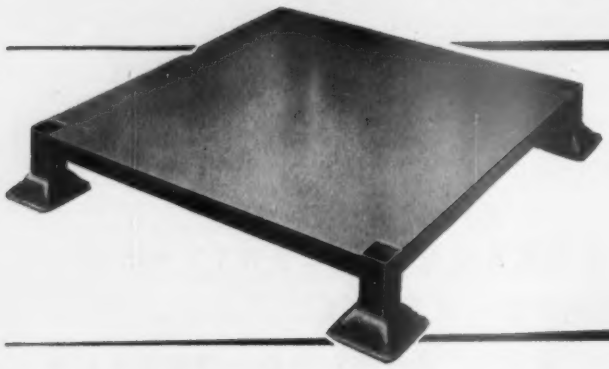

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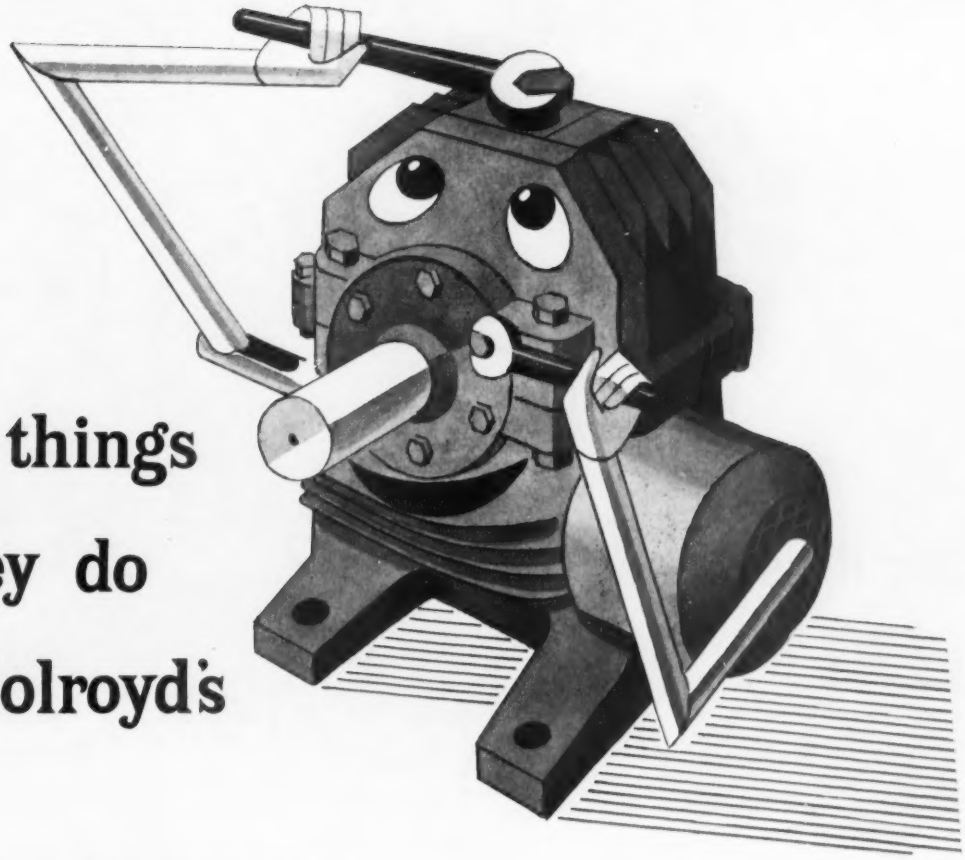

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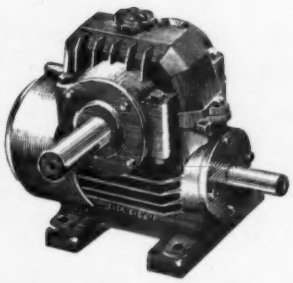

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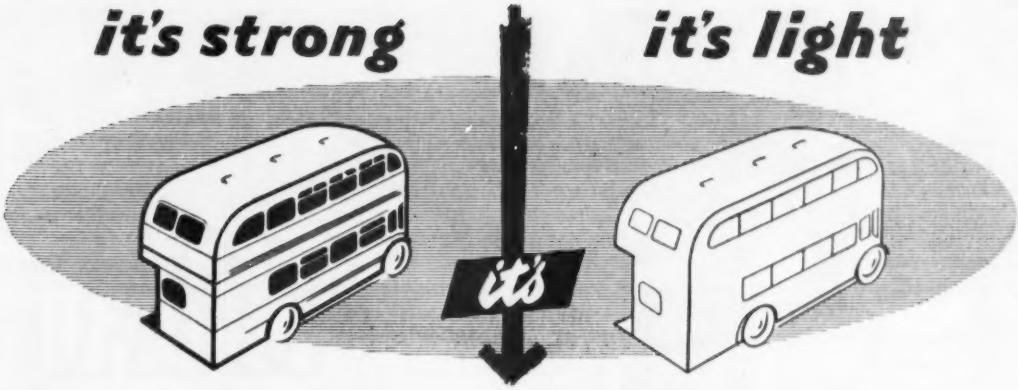

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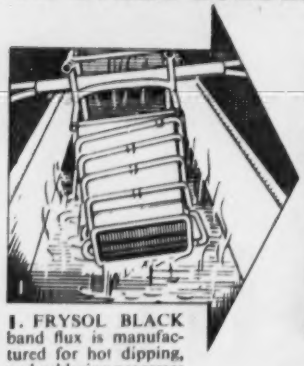

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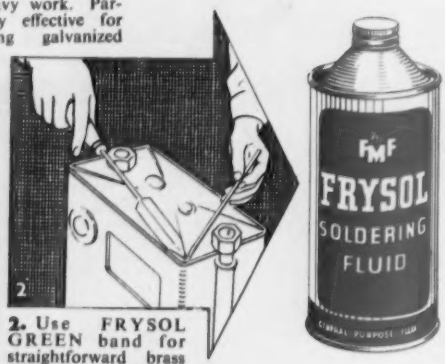

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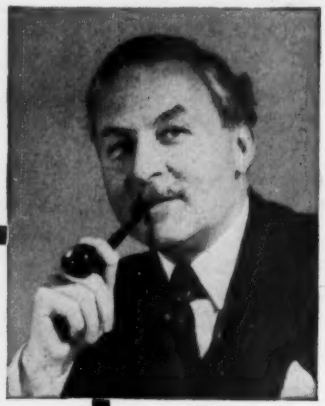

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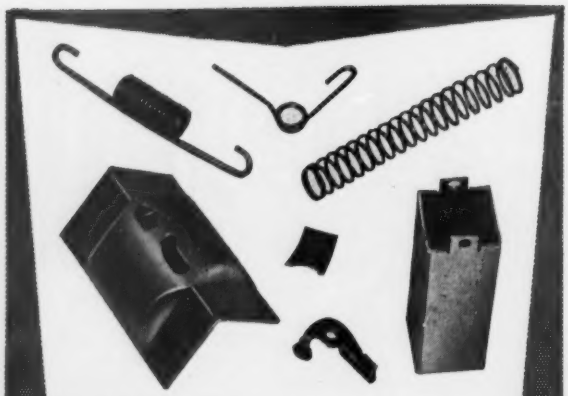

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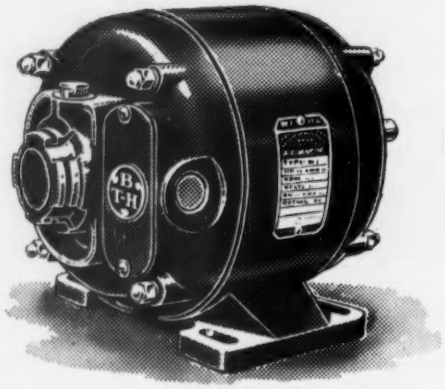

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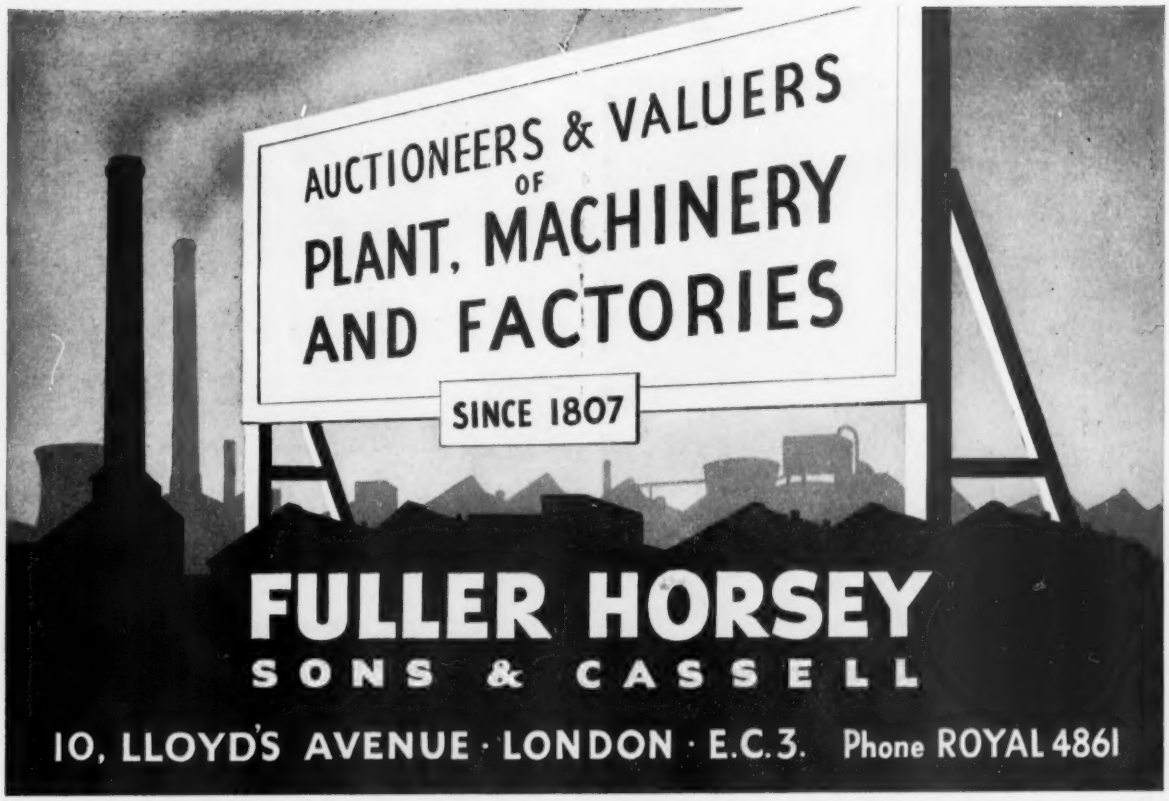

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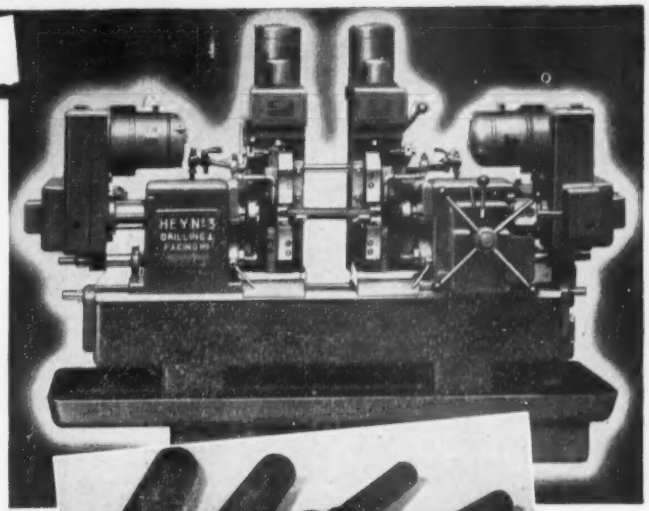

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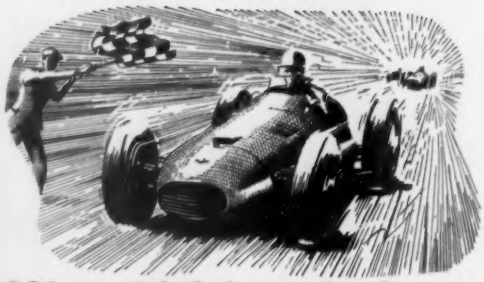

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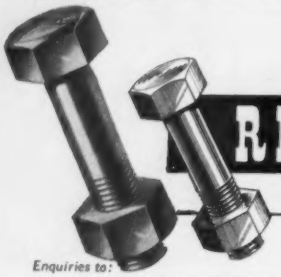

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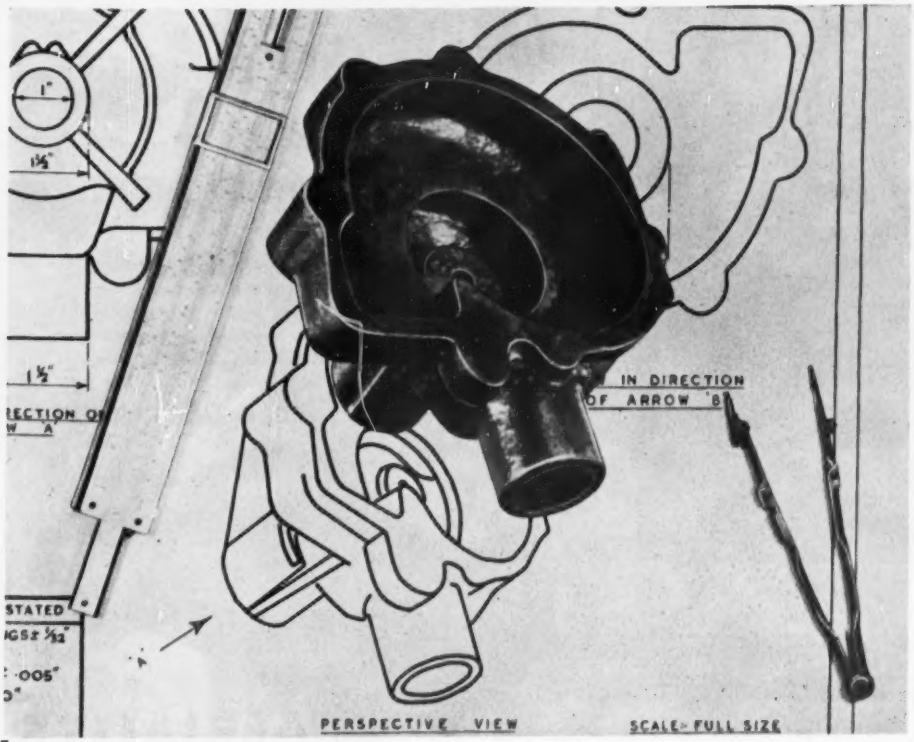

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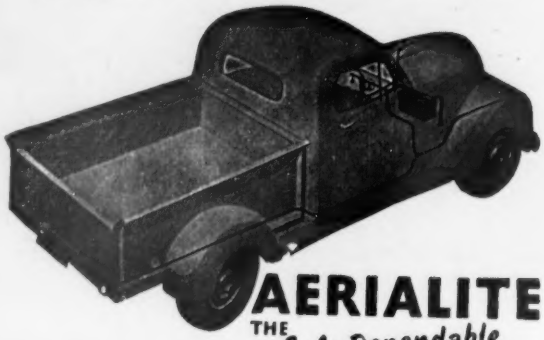

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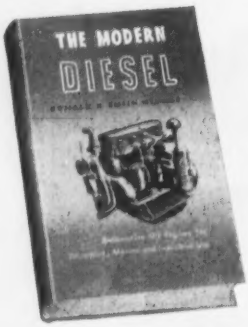

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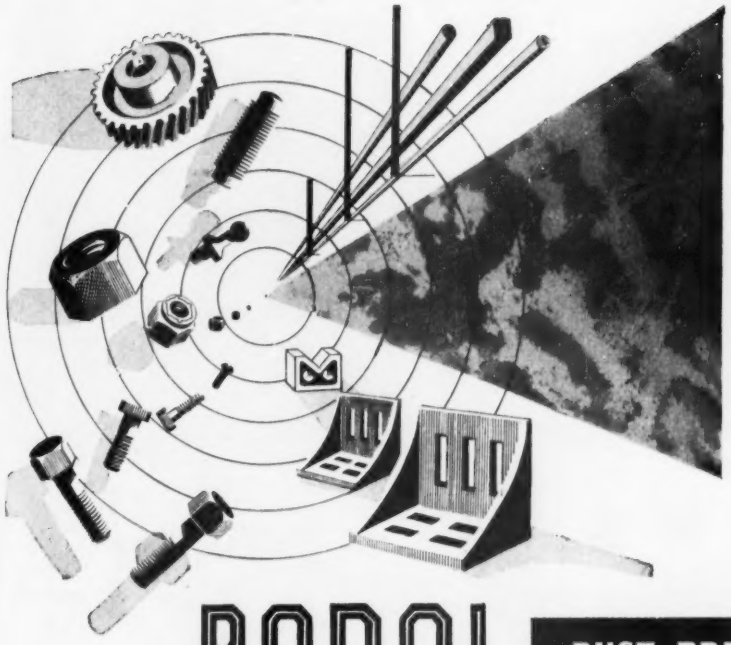

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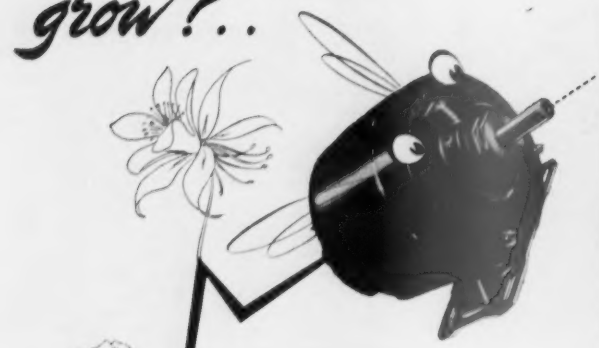

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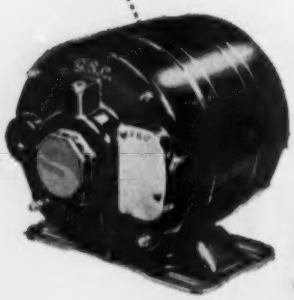

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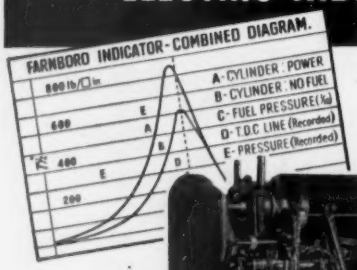

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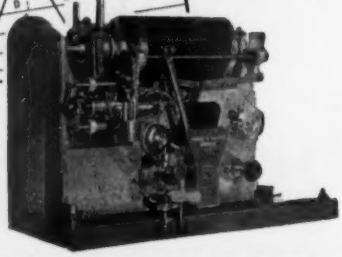

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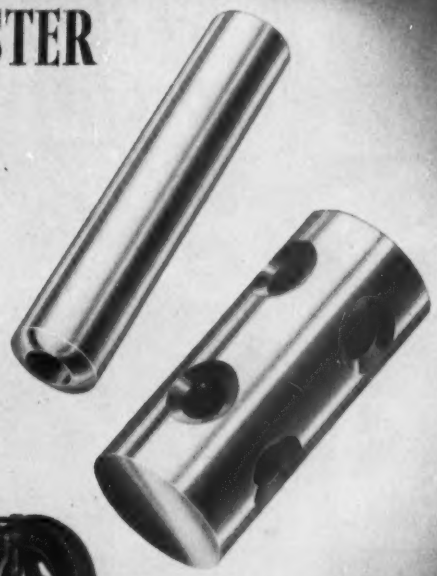

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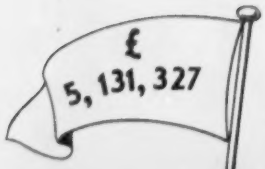

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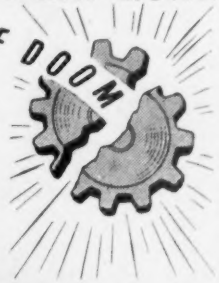

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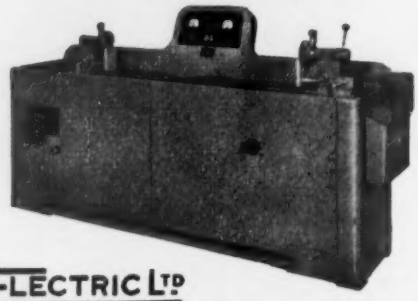

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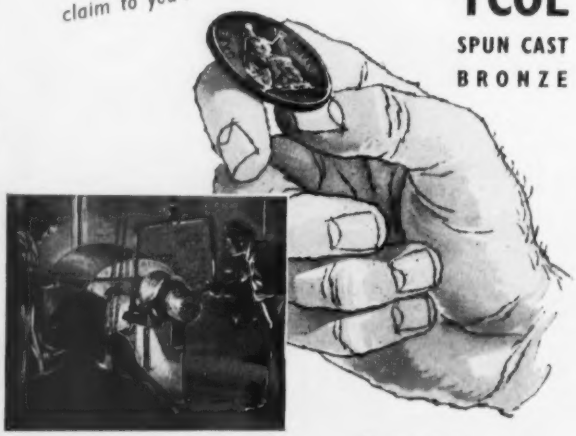

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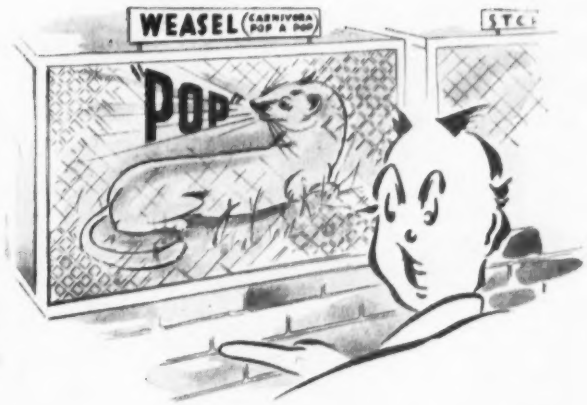

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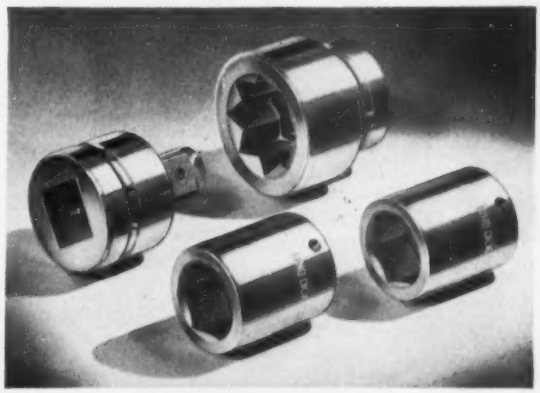

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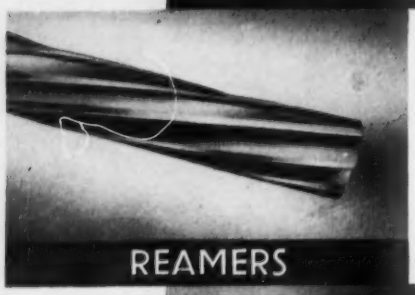

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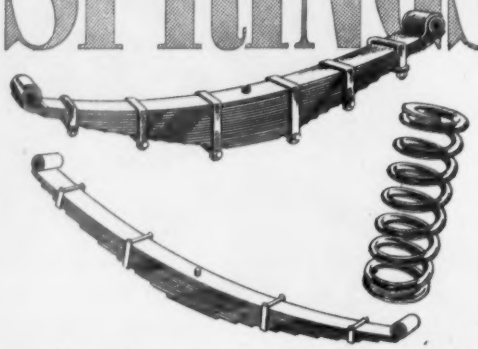

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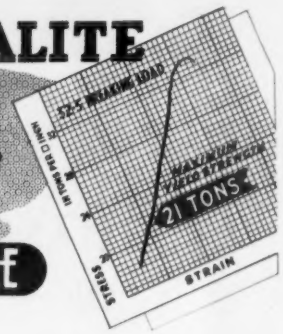

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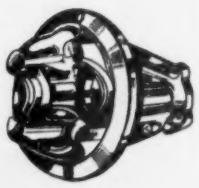

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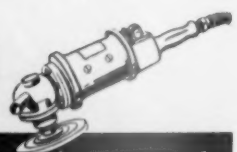

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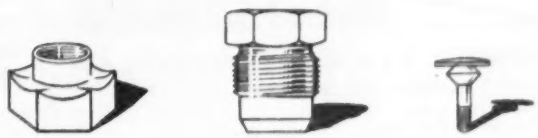

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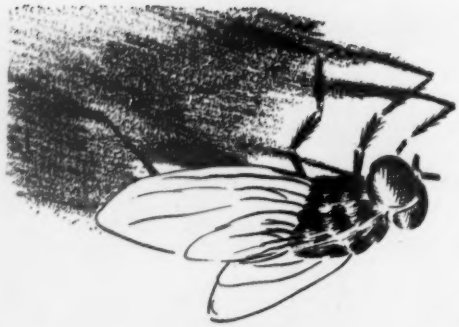

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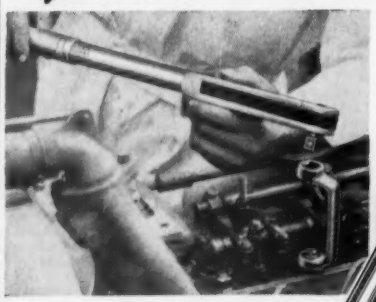

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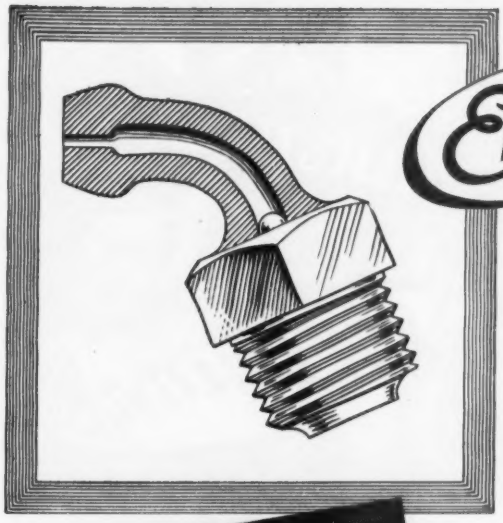

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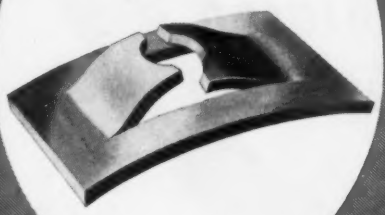

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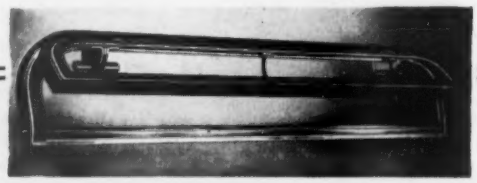

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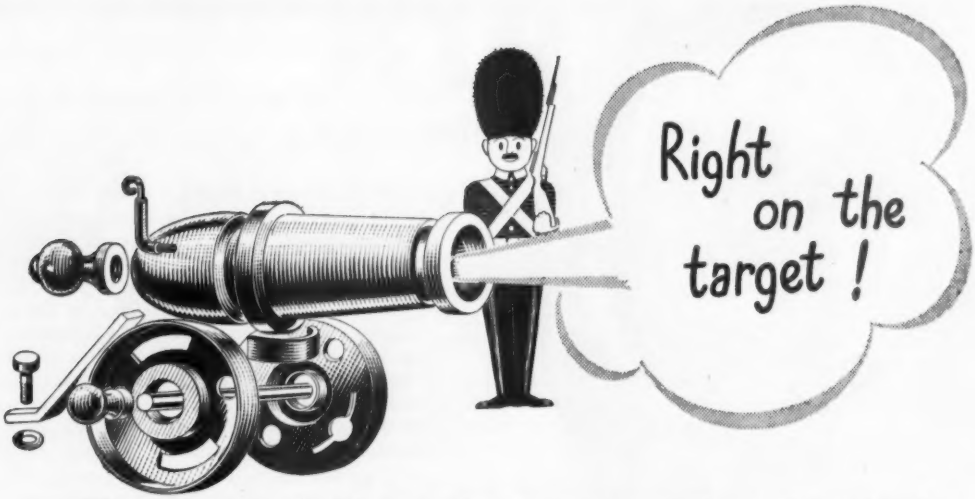

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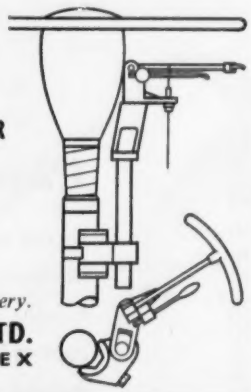

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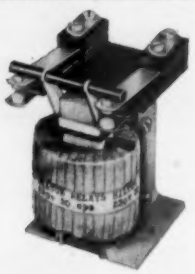

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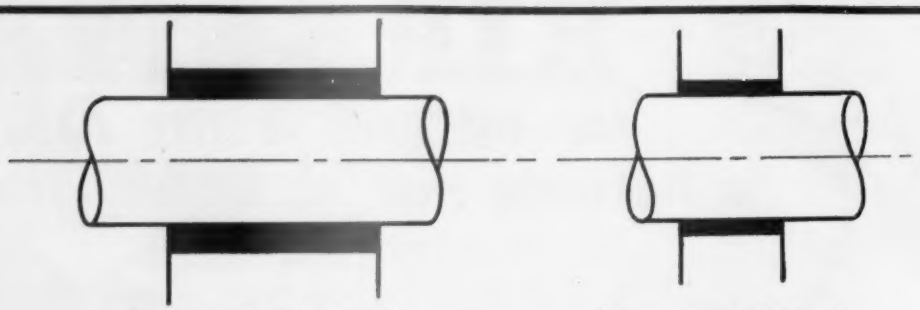

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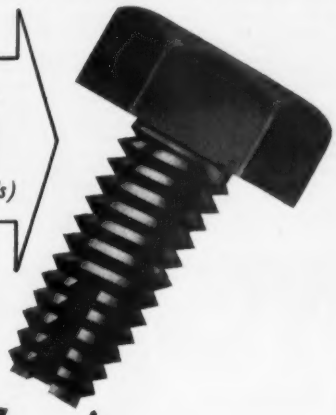

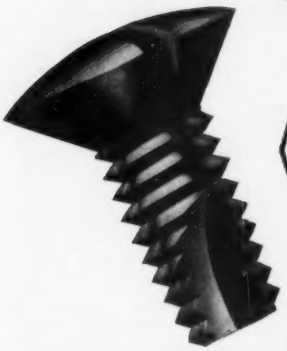

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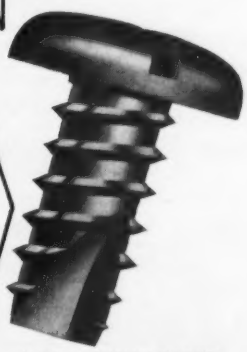

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PATENTS

NOTICE is hereby given that William Wilson Hamill seeks leave to amend the Complete Specification of the Letters Patent No. 659102 for an invention entitled "Means for Controlling, Regulating or Governing Speed of Rotation."

PARTICULARS of the proposed amendments were set forth in the Official Journal (Patents) No. 3409 dated 16th June 1954.

ANY person may give Notice of Opposition to the amendment by leaving Patents Form No. 36 at the Patent Office, 25, Southampton Buildings, London, W.C.2, on or before 16th July 1954.—COMPTROLLER-GENERAL. [4849

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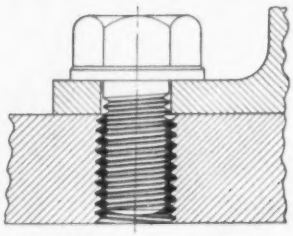

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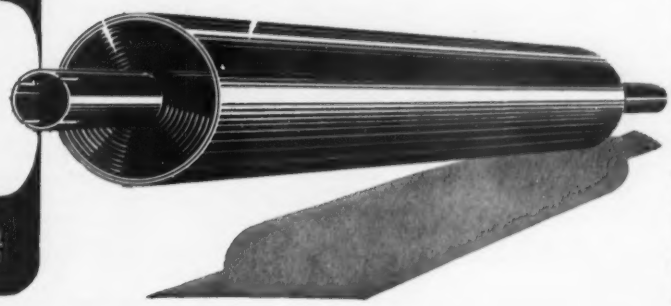

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Printed in Great Britain for the Publishers, ILIFFE & SONS LTD., Dorset House, Stamford Street, London, S.E.1, by James Cond Ltd., Charlotte Street, Birmingham 3. "Automobile Engineer" can be obtained abroad from the following: AUSTRALIA & NEW ZEALAND: Gordon & Gotch Ltd. INDIA: A. H. Wheeler & Co. CANADA: The Wm. Dawson Subscription Service Ltd. Gordon & Gotch Ltd. SOUTH AFRICA: Central News Agency Ltd, Wm. Dawson & Sons (S.A.) Ltd. UNITED STATES: The International News Co. Entered as Second Class Matter at the New York, U.S.A., Post Offices.

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